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U. S. S. *ASHEVILLE.*

DESCRIPTION AND OFFICIAL TRIALS.

BY LIEUT. COMDR. H. G. DONALD, U. S. NAVY, MEMBER.

HISTORICAL DATA.

Gunboat No. 21, the *Asheville,* is one of two vessels authorized by act of Congress dated August 29, 1916. The other vessel, Gunboat No. 22 is to be christened *Tulsa,* and will be a duplicate of the *Asheville.*

The *Asheville* was built at the Navy Yard Charleston, South Carolina, on order placed August 20, 1916, with contract date of completion December 31, 1918. The keel was laid June 9, 1917, and the vessel was launched July 4, 1918, being christened by Miss Alyne Reynolds. The construction was delayed because of the War but final delivery and commissioning took place July 17, 1920.

On September 27, 1920, a trial was held under the supervision of the Navy Yard and the vessel joined the Special Service Squadron in Central American Waters. The ship returned to the United States in May, 1921. After a short overhaul at Charleston and docking at Norfolk, standardization, full power, backing and steering trials were conducted July 7, 1921, by the Board of Inspection and Survey off Provincetown, Mass. Details of these trials appear later in this article.

The cost of the *Asheville* was $1,287,911 divided as follows: Hull $888,771, machinery $399,140. The original limit of cost was $1,100,000 which was increased to $1,250,000 June 4, 1920.

GENERAL.

The *Asheville* is a single screw gunboat, built of steel. The main and second decks are flush, and there is a platform deck forward and aft of the machinery spaces. The ship is designed for a speed of 12 knots at 1575 tons displacement. The propelling machinery consists of one set of two turbines, compounded and driving the one main shaft through single reduction gearing. There are three Bureau Engineering Express type boilers which, on the *Asheville* only, may use either all oil or all coal fuel, the conversion from one to the other being within the capacity of the ship's force. By Department's order the boilers of the *Tulsa* are to be fitted only for oil burning.

HULL DATA.

General Dimensions:

Length over all, feet and inches	241–2
Length between perpendiculars, feet and inches	225–0
Breadth, molded, feet and inches	41–1
Breadth, extreme, feet and inches	41–2½
Depth of side, least molded, feet and inches	23–9½
Freeboard at stem, feet and inches	16–4⅛

Freeboard at stern, feet and inches	14–11⅞
Dead rise, inches	8
Molded depth of inner bottom at center line, inches	26
Number of frames	74
Frame spacing, inches	36
Draft forward (1592 tons)	10–2⅜
Draft aft (1592 tons)	12–5⅝
Calculated Data (for 11 feet 4 inches water line):	
Tons per inch	14.32
Displacement, inches	1,592
Area of midship section, square feet	437
Area of normal water plane, square feet	6,025
Wetted surface, square feet	10,050
Coefficient block	.53
Coefficient prismatic	.57
Coefficient midship	.93
Coefficient water line	.65
Area of rudder, square feet	63
Center of buoyancy above bottom of keel, feet and inches	6–4¼
Center of buoyancy forward of middle perpendicular, feet	.73
Traverse metacenter above center of buoyancy, feet and inches	17–5½
Longitudinal metacenter above center of buoyancy, feet and in.	241–7½
Center of gravity of water line abaft middle perpendicular, feet and inches	3–½
Molded base above bottom of keel, inches	13/16
Center of gravity of full-load water line abaft middle perpendicular	3–10¼
Midship section forward of frame 37, inches	6
Center of gravity above bottom of keel, actual, feet	15.55
Traverse G. M. full load (12 feet 4¾ inches W. L.), feet	2.05
Center of gravity above bottom of keel at full load, feet	14.99
Longitudinal G. M. full load, feet	224.98
Heights above 11 feet 4 inches water line:	
Truck light on foremast, feet and inches	129–6½
Truck light on mainmast, feet and inches	129–2
Crow's nest on foremast, feet and inches	90–6
Bridge at center, feet and inches	26–6
Wireless antenna	122–4

Conditions of Loading:—The conditions of loading, ready for service in every respect, with full complement of officers and crew with their effects, and consumable load, is tabulated below for normal, full and emergency conditions. In the design of

the vessel the mean draft corresponding to the designer's water line—viz., 11 feet, 4 inches—contemplates the condition of loading given under the heading "Normal."

Item	Load Condition		
	Normal, tons	Full, tons	Emergency, tons
Hull	655.12	655.12	655.12
Hull fittings	175.63	175.63	175.63
Protection	8.46	8.64	8.64
Engineering	132.70	132.70	132.70
Reserve feed water	5.33	8.00	45.66
Battery	21.68	21.68	21.68
Ammunition	54.06	54.06	54.06
Equipment	76.54	76.54	76.54
Outfit	129.70	162.51	162.51
Coal	300.00	450.00	534.20
Officers, crew and effects	15.78	15.78	15.78
Oil fuel	.....	.....	146.10
Total	1,575.00	1,760.48	2,028.44

Complement and Quartering:—

Officers	13
MEN.	
Seamen Branch, C.P.O.	4
Seamen Branch, other P.O.'s	19
Seamen Branch, non rated	50
Deck artificers	4
Engineer force, C.P.O.	4
Engineer force, other P.O.'s	17
Engineer force, non rated	24
Yeomen	8
Hospital Corps	2
Bugler	1
Commissary branch	5
Messmen	11
Total, men	162

The officers are quartered forward on the second deck, the captain's cabin being above on the main deck. There are twelve officers' staterooms. The chief petty officers' quarters are aft on the platform deck. One crew space of 3,323 square feet floor area is aft on the second deck, and one of 798 square feet area is forward on the platform deck. The galley, bake-shop and laundry are located on the main deck. The sick bay is on the platform deck forward and has six hinged berths.

Draft Gauges:—Besides the usual outside draft marks there are two internal draft gages.

Armament:—The battery is composed of three 4-inch, 50-caliber, rapid-fire guns located on the main deck and an auxiliary battery of two 3-pounders, two 1-pounders, four 30-caliber machine guns and two 3-inch field guns. An armored conning tower is located forward on the main deck; and the 4-inch guns are provided with bullet-proof shields. Stowage space for an ample supply of ammunition is provided in six magazines.

Anchor Gear:—Two bower anchors, each 5,000 pounds, are stowed in the hawse pipes to port and starboard. Each of these anchors has 135 fathoms of 1¾-inch Government standard chain. Other anchors are one 3,000-pound stockless stern anchor and two kedge anchors, one 750 pounds and one 400 pounds.

An anchor windlass on the main deck forward, engine on platform deck, is capable of raising both anchors simultaneously at the rate of 6 fathoms per minute. The engine is vertical, double cylinder, steam operated.

Deck Winches:—The vessel has two double-headed reversible steam winches, one forward and one aft on the main deck. The capacity of each is 4,000 pounds at 50 feet per minute and 1,000 pounds at 125 feet per minute; gears being provided for the two speeds.

Ash Handling:—Ashes are hoisted through the port fire room ventilator by a double-cylinder Hyde ash hoisting engine.

Boats:—The following boats are provided:

SIZE AND TYPE.

	Number
36-foot motor sailing launch	1
30-foot motor sailing launch	2
30-foot steamer	1
28-foot whaleboat	2
16-foot dinghy	2
14-foot Wherry	1

All boats are stowed on the superstructure deck and are hoisted by hand.

Fuel Stowage:—There are the following coal bunkers on the *Asheville* only and fuel oil tanks on both *Asheville* and *Tulsa:*

Coal Bunkers.

No. of bunker	Capacity in cu. ft. to bottom of beams	Capacity in tons at 42 cu. ft./ton
B 102	1,377	32.8
B 103	1,377	32.8
C 101	7,547	179.7
Total	10,301	245.3

Fuel Oil Tanks (95 per cent capacity).

Tank	Cubic Feet	Gallons at 7.481 gal./cu. ft.	Tons
A 116	6,133.3	46,000	153.3
A 117	6,133.3	46,000	153.3
A 3	524.78	3,926	13.12
A 4	524.78	3,926	13.12
A 5	568.99	4,257	14.23
A 6	568.99	4,257	14.23
A 7	635.56	4,755	15.89
A 8	635.56	4,755	15.89
C–1	420.13	3,143	10.50
C–2	420.13	3,143	10.50
C–3	679.67	5,085	17.00
C–4	679.67	5,085	17.00
D–1	316.49	2,368	7.91
Total	18,241.25	136,700	455.99

Coal is taken aboard by four coaling booms, two on each side of the main deck, the whips being handled by the above described winches.

The fuel oil tanks are filled through four 4-inch filling connections, two forward and two aft on the main deck, one of each group being on each side of the ship. Both connections of each group are connected, as also are the two groups. The forward filling lines connect to a distributing manifold in the fire room and the after lines to a manifold in the engine room. The manifolds have connections to each tank, common line being used for filling and suction, and also connections for suction and discharge are made to the two fuel oil booster pumps. These pumps have a sea connection for use when washing out the tanks. High suctions are provided for tanks A-8 and C-1.

Steering Gear:—The vessel is fitted with one 5 × 5 inch American right and left hand screw-gear type steering engine attached directly to the rudder crosshead, operated by a horizontal engine, having two 5 × 5 inch cylinders operating the screw shaft through a worm wheel. The steam steering stations are on the bridge and in the conning tower. An emergency hand steering station is located on the main deck aft, tackles being connected to the spare tiller. The power gear is capable of throwing the rudder from hard over to hard over in 15 seconds when the ship is making 12 knots.

Drainage System:—The drainage system consists of a main drain and a secondary drain, each entirely independent. The main drain, 7 inches diameter, extends on the port side from after end of boiler room to the main circulating pump in engine room, suction to boiler room and engine room, with deck reach rods. The secondary drain, 3 inches and 4 inches diameter, extends on the starboard side of the ship from frame 13 to the after trimming tank, branches take suction from the following compartments Fore peak tank, A-102, pump room, double bottoms A-1 and A-2, forward ammunition handling room, bunkers A-116 and A-117, boiler room, engine room,

bunkers C-101, B-102 and B-103, shaft alley, double bottom D-2, and the after peak tank. Suction is put on the secondary drain with the two fire and bilge pumps, one in the boiler room and one in the engine room. All important valves can be operated from a deck above the compartment in which the valve is located. All pipes are galvanized steel tubing.

Fire Main:—The fire main consists of a single line 5-inch outside diameter steel tubing, lead lined; running from frames 19-20 platform deck, port, to frame 56-57, platform deck, starboard, crossing over at frame 22. The outer tubing (steel) has perforations 1/4 inch diameter spaced 12 inches apart, one row top and bottom for locating leaks in lead lining. All branches are steel tubing, lead lined. Pressure is supplied by the two fire and bilge pumps in the boiler and engine rooms. The fire main supplies 11 fire plugs located as follows: crew's space A-205, C.P.O. quarters D-201, crew's space C-301, main deck at frames 16-17 port and starboard 30-31, 49-50 port and starboard, 66-67. Direct connections are provided as follows: forward and after magazine sprinkling system, distilling condenser, ice machine, lubricating oil, circulating water and stern tube. The peak tanks are flooded through manholes by fire hose.

Stationary Foam Fire Extinguishers are located in the fire room and engine room.

Flushing System:—The flushing system consists of a 4-inch outside diameter pipe line, running along the port side of the second deck, with branches supplying the following spaces with their fixtures. Wardroom water closet (A-305); wardroom urinal (A-303); sick bay; captain's bath; wash-deck hopper in C-301; bakery; galley; potato peeler; laundry; firemen's wash room; general mess pantry; crew's water-closet and wash room. Pressure is supplied by two Terry steam turbine driven, Worthington centrifugal pumps in the engine room (the *Tulsa* is to have only one pump). Each pump has a

capacity of 30,000 pounds water per hour against a 60-foot head. A relief valve is set at 35 pounds gauge. All piping is steel tubing, galvanized, lead lined.

Fresh Water System:—There are two ship's fresh water tanks A-103 and 104 having a total capacity of 39.84 tons or 10,728 gallons. The capacity of the reserve feed tanks is 45.66 tons or 12,282 gallons. Deck filling connections are located on the shell of the ship, port and starboard and on the main deck at frames 30-31 and 40-41, port and starboard. Two electrically driven centrifugal pumps, capacity of each 70 gallons per minute at 40-foot discharge head, take suction from the above tanks and discharge through a 2½-inch main terminating in two 50-gallon gravity tanks located under the main deck at frames 71-73 for crew's and chief petty officers' wash rooms and a 75-gallon scuttle butt. Branches from the main supply the following: Wardroom bathroom; each wardroom stateroom; sick bay dressing room and bathroom; captain's bath and pantry; at frame 45 starboard side for filling steamer; galley, laundry; bakery; potato peeler, general mess pantry. A branch also supplies a 50-gallon gravity tank which feeds the firemen's washroom, engineers' office and chief petty officers' quarters. The auxiliary feed pump is connected by a 2½-inch pipe to the fresh water main for filling reserve feed tanks from deck. The evaporator fresh water pump discharges to the fresh water main; near the ship's tanks. All pipes are galvanized iron.

Ventilation and Heating:—Artificial ventilation is accomplished by nine systems each having a motor driven fan. The motors and fans were supplied by the Ilg Electric Ventilating Co., Chicago. For each system, the power of the motor, capacity of fan, and compartments served are as follows:

System No.	Type	Motor Horsepower	Capacity (cu. ft./min.)	Serves Compartments
1	Supply	2.5	2,500	A-206-319-321 C-301
2	Supply	2.5	2,500	A-205-319-320-322, C-301

3	Exhaust	.38	600	A–206
4	Supply	3.4	4,000	C–102–301, D–201
5	Supply	3.4	4,000	C–102–301, D–102m–103m –101–107–202
6	Exhaust	2.5	2,500	C–301, D–304–305–306
7	Supply	2.5	2,500	A–105–108m–111m–113m–114–202–203–204–303–307 to 319 incl.
8	Supply	2.5	2,500	C–102
9	Supply	2.5	2,500	C–102

In quarters, crew spaces, etc., that require heating, the air, before delivery is passed through heaters supplied by B. F. Sturtevant Co. Humidity is controlled by a hand valve admission of steam. By-pass temperature control dampers permit of fine adjustment of temperature.

Galley, Bake Shop, Potato Peeling Room and General Mess Pantry:—The galley is equipped with the following:—Four 3½-foot Navy standard coal burning ranges, two 60-gallon steam jacketed kettles, one 60-gallon coffee urn, one 300-pound/hour motor driven meat grinder, one 60-quart motor driven kitchen and coke machine.

The bake shop equipment is as follows:—One (proofer) steam box No. 1 USN, one motor driven dough mixer, two No. 2 USN bake oven.

The potato peeling room has one motor driven 500 pound/hour potato peeler and one potato grinder.

In the general mess pantry there is a motor driven dish washing machine with capacity for 6,000 pieces/hour and a butter slicer.

A 20-quart hand ice cream freezer is installed near the cold storage rooms.

Laundry:—The laundry is equipped with a 36-inch, 3 H.P., Thor washing machine, one 5-foot ironing board and one two-compartment tub. Space in the uptake is utilized as a drying room.

DESCRIPTION OF MACHINERY.

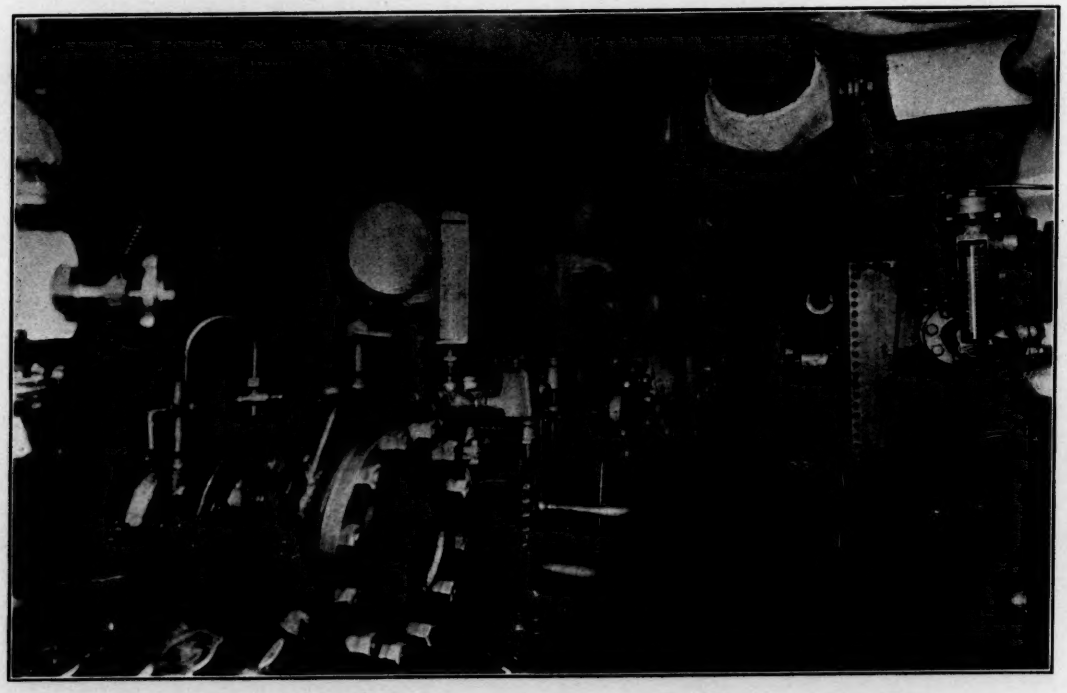
General:—The main propelling machinery consists of one set of single reduction, mechanical geared, turbines of the Parsons or reaction type, placed in one water tight compartment. A single shaft is driven through gearing by one high-pressure and one low-pressure turbine of combined shaft horsepower of 800 at 127 R.P.M. In the casing of the low-pressure turbine, there is fitted an astern turbine capable of developing 50 per cent of the full power ahead. The general arrangement of the machinery is as shown on plate. The boiler working pressure was originally 200 pounds gauge, but was later raised to 215 because of the fact that the turbo-generators require steam 200 pounds gauge pressure at full load. The steam pressure at the high pressure turbine is 140 pounds gauge and at the astern turbine was originally 50 pounds gauge, but has been raised to 75 pounds on account of trouble had in backing as detailed later. The main auxiliaries include main condenser, of 1,165 square feet C.S., an independent reciprocating main air pump, an air ejector for the condenser, and a centrifugal circulating pump.

There are three small tube water-tube boilers of the "Express" type in one water tight compartment. The boilers of the *Asheville*, only, are fitted for burning either coal or oil, the necessary gear for each being carried on board. The *Tulsa's* boilers will burn oil only. The total grate surface is 295.5 square feet, total furnace volume (for oil) 1,326.75 cubic feet and total heating surface is 11,850 square feet. The forced draft system originally consisted of a single turbine-driven blower discharging into a closed fire room and later an additional blower was added. There is one smoke pipe. A complete system for burning fuel oil is provided. The auxiliary machinery installation consists of one condenser, 3 feed pumps, 2 fire and bilge pumps, 3 fuel oil pumps, a fourth being added after commissioning, refrigerating plant of 2

tons per day capacity, 6,000 gallons per day distilling plant, 1 air compressor for general service, and a small machine shop.

Main Turbines:—The main turbines are of the Parsons type, one high-pressure turbine in a single casing, compounded with and set parallel to a low-pressure turbine. In the casing of the latter turbine at the forward end, and mounted on the same spindle is the astern turbine. The plans for the turbines were purchased from the Parsons Marine Steam Turbine Company of New York City and the turbines were constructed at the Charleston Navy Yard, where the vesel was built. None of the turbines on the *Asheville* has a dummy cylinder as is customarily found in this type of turbine. It was estimated that on account of the small size of the turbines the end thrust could be easily handled by the turbine thrust bearings without excessive friction loss.

On shop test of the *Asheville's* turbines and also after installation in the ship, but before delivery, it was found to be impossible to start the turbines astern without using a steam chest pressure much in excess of the designed figure (50 pounds). At times a pressure of 150 pounds gage was required. This difficulty in backing was particularly noticeable if the turbines were started astern from rest, that is if the turbines had not immediately before been going ahead. All possible sources of excessive friction, and even possible reversal of blades was investigated most carefully without result. Finally, it was decided that the excessive starting friction must have been due to the end steam thrust binding the shoes of the turbine thrust bearing (which is of the segmental type) on the thrust collar. Attempts were made to relieve this friction by first easing off a few thousandths of an inch on the leading edge of the shoes and then by installing buttons on the back of the shoes to prevent them, when at rest, from becoming exactly parallel with the thrust collar. Neither method was successful. As installed, the support for each shoe of the segmental thrust was supplied by a steel button let in at the geometrical center



U. S. S. "Asheville," View of Engine Room, For'd End Looking to Port.

of the back of the shoe. Finally, a set of shoes was installed in the low-pressure turbines thrust bearing on the side taking the astern thrust, having the supporting button on the back of each shoe located at the mean center of oil pressure when the shaft is in motion. This position is approximately one-third of the circumferential length of the shoe from the off, or leaving, edge of the shoe. This same alteration has been made to the low pressure trust of the *Tulsa.*

Backing tests at once showed remarkable improvement, the same vacuum being maintained in the main condenser as on previous tests. The astern turbine could be started with a pressure lower than 50 pounds gauge in the steam chest. At a later date, the maximum steam chest pressure of the astern turbine was raised to 75 pounds gage as this pressure is required to start the turbines astern quickly (15 seconds) when the ship is making considerable speed ahead. In order to guard against this trouble on the *Tulsa,* a dummy has been fitted in the low-pressure turbine in a manner similar to that on a number of Shipping Board vessels—that is, by cutting out the inner steam gland at the forward end and installing a dummy cylinder in its place. On the *Asheville,* with a 50-pound gauge steam pressure on the first stage astern steam space and a 20-inch vacuum at exhaust, the unbalanced thrust would be 5,986 pounds and the pressure per square inch on thrust shoes would be 163 pounds. The total unbalanced force is given by the formula, $101.4\ P_o + 121.0\ P$, where P_o is the absolute pressure in steam space at entry into first stage blading and P is the exhaust pressure, absolute. The area of shoes on one side of the thrust is 37 square inches. The details of the turbines are as listed in Table I. All blading is of the reaction type.

Reduction Gearing:—The turbine drives the single main shaft through a reduction gear; the ratio of speed of both the high pressure and low pressure turbines and the main shaft

TABLE I.

TURBINE DATA.

Expansions	Ahead: Diameter of Cylinder H. P. inches	Ahead: Diameter of Cylinder L. P. inches	Ahead: Length of Cylinder this diameter H. P. inches	Ahead: Length of Cylinder this diameter L. P. inches	Ahead: Rows of Blades H. P.	Ahead: Rows of Blades L. P.	Ahead: Pitch of Blades H. P. inches	Ahead: Pitch of Blades L. P. inches	Astern: Diameter Cylinder inches	Astern: Length	Astern: Rows of Blades	Astern: Length Blades inches	Astern: Pitch Blades inches
1	11-$\frac{3}{4}$	23	11-$\frac{1}{8}$	4-$\frac{5}{16}$	12	3	$\frac{1}{4}$	$\frac{1}{4}$	14-$\frac{3}{4}$	5	4	$\frac{3}{4}$	$\frac{1}{4}$
2	12-$\frac{1}{16}$	23-$\frac{5}{8}$	10-$\frac{5}{8}$	4-$\frac{1}{16}$	12	3	$\frac{1}{4}$	$\frac{1}{4}$	15-$\frac{1}{16}$	4$\frac{1}{2}$	4	$\frac{7}{16}$	$\frac{1}{4}$
3	12-$\frac{1}{2}$	25	10-$\frac{3}{4}$	4-$\frac{1}{16}$	12	3	$\frac{1}{4}$	$\frac{1}{4}$	15-$\frac{1}{8}$	3$\frac{1}{2}$	3	$\frac{1}{2}$	$\frac{1}{4}$
4	15-$\frac{3}{8}$	26-$\frac{1}{2}$	7-$\frac{9}{16}$	4-$\frac{1}{16}$	7	3	$\frac{1}{4}$	$\frac{1}{4}$	16	4$\frac{1}{2}$	3	1	$\frac{1}{4}$
5	16	27-$\frac{1}{2}$	8-$\frac{5}{16}$	2-$\frac{1}{16}$	7	2	$\frac{1}{4}$	$\frac{1}{4}$	16-$\frac{3}{4}$	4$\frac{1}{2}$	3	1-$\frac{3}{16}$	$\frac{1}{4}$
6	16-$\frac{1}{4}$	29-$\frac{1}{4}$	9-$\frac{1}{8}$	3-$\frac{1}{8}$	7	2	$\frac{1}{4}$	$\frac{1}{4}$	17-$\frac{1}{2}$	3$\frac{1}{2}$	2	1-$\frac{3}{16}$	$\frac{1}{4}$
7	...	29-$\frac{1}{2}$	...	1-$\frac{5}{16}$	...	1	...	$\frac{1}{4}$	17-$\frac{1}{2}$	1$\frac{1}{2}$	1	1-$\frac{3}{16}$	$\frac{1}{4}$
8	...	...	...	...	...	...	...	...	17-$\frac{1}{2}$	1$\frac{1}{4}$	1	1-$\frac{1}{4}$	$\frac{1}{4}$

	H. P.	L. P.	Astern.
Length of Rotor Drums, feet and inches	2-10$\frac{1}{2}$	2-6$\frac{3}{8}$	2-6
Diameter of Rotor Drums, inches	11	21	14
Material of Casing	Cast iron	Cast iron	Cast iron

being 28 to 1. The drive of both pinions is on the horizontal plane passing through the gear shaft. The turbine pinions are connected to the turbines by "star" or jaw type flexible couplings. The design of the gearing is by Parsons Marine Steam Turbine Company and is typical, the pinions being each supported by three rigid bearings.

PARTICULARS OF GEARING.

	H. P.		L. P.	
	Wheel	Pinion	Wheel	Pinion
Number of teeth	97	21	233	39
Pitch circle diameter inches	23.767	5.146	58.277	9.755
Diameter over teeth inches	24.007	5.386	58.517	9.995
R. P. M.	...	3500	127	...
Circular pitch	.7697	.7697	.7857	.7857
Spiral angle	40°-45.9′	40°-45.9′	42°-6.1′	42°-6.1′

The main gear spider is of cast iron and the shrouding in which the teeth are cut is forged steel shrunk on the spider. The pinions and sleeves over the pinions are nickel steel. All gearing is double helical.

The gearing has operated very satisfactorily but is quite noisy at speeds over 10 knots.

Control:—Balanced throttle valves are provided one for admitting steam to the high pressure turbine for ahead running and one for steam to the astern turbine for backing. The two throttles are interconnected and operated through a lever by one wheel. In the closed position the throttle discs are held on their seats by steam pressure, and the pilot valves are held closed by springs. A connection is provided for by-passing the high pressure exhaust around the low pressure turbine direct to the main condenser in case the low pressure turbine becomes disabled. This line is ordinarily blank flanged.

Shafting and Bearings:—Data on shafting and bearings are as follows:

H.P.	L.P. & Ast.
Rotor shaft, Material Cl. Ac. Steel Forg...	Cl. Ac. Steel Forg.
Rotor shaft, Length, feet and inches, 8–9⅛	8–5⅝
Rotor shaft, Diameter at journals, inches, 4½.....	4½
Rotor shaft, Bearings length,	
white metal, inches, 4.....	4
overall, inches, 5½.....	5½
Rotor shaft, Bearings diameter,	
white metal, inches, 4½	4½
brass, inches, 4$\frac{17}{32}$	4$\frac{17}{32}$
Turbine thrust, type and size, inches, Kingsbury 9	Kingsbury 9
Main thrust, type and size, inches, Kingsbury 14	Kingsbury 14

	Thrust Shaft	Line Shaft	Propeller Shaft
Bearings, No.....	2	2	1 1
Bearings, length, inches.....	20	20	24 33
Bearings, diameter, inches	8	8	9½ 9½
Shaft, length, feet and inches	22–3	20–0	21–7⅛
Shaft, diameter, inches.....	8	8	8½
Shaft, bore, inches	4	4	... 4
Material	Cl. B Steel Forg.	Cl. B Steel Forg.	Cl. B Steel Forg.

The turbine thrust bearings are forward of the turbines and are unusually large because of the fact that the turbines have no dummies and the full steam thrust on the end of the rotor must be provided for. The main thrust bearing is aft of the gear casing.

Propeller:—There is one right handed true screw propeller. It is of solid construction of manganese bronze.

The propeller originally installed was reported by the *Asheville* to be not entirely satisfactory. An investigation of the design disclosed the fact that the propeller would be expected to function badly in heavy weather and accordingly a new propeller with fan shaped blades was designed and installed previous to the trials. Comparison of the two wheels is as follows:

	New	Old
No. of blades	3	3
Diameter, feet and inches	10–0	9–8
Pitch, feet and inches	10–9	10–11
Pitch/Diameter	1.075	...
Area, projected	25.59	22.53
Area, helicoidal	30.84	...
Area, disc	78.54	...
PA/DA	.321	.31
HA/DA	.392	...
PA/HA	.829	...
Height lower tip over keel, inches	9	...
Immersion upper tip, 12 feet, 7¾ inches, draft aft, inches	22¾	...
R.P.M., Designed	126	127
S.H.P.	808	...
Speed, Knots	12	...

The new propeller functioned very well on the trials. The R.P.M. versus speed curve is as shown in Plate I. The R.P.M. for 12 knots from the curve would have been 133.4. The discrepancy between this and the designed figure of 126 is accounted for by increased displacement, 132 tons over, and the poor trim existing at the time of the standardization trial, which was 3 feet 3½ inches down by the stern.

Turning Gear:—The main turbine turning gear is hand operated.

Torsion Meters:—A Gary-Cummings torsion meter is installed on the main shaft of both the *Asheville* and the *Tulsa*. The shaft calibration data are as follows for the *Asheville* only:—

Length of shaft calibration, feet and inches	6-0
Foot pounds for one inch torque on 96-inch radius	15,650
Length of shaft which torsion meter records, feet and inches	19-7
Torsion meter constant	2.98

Engine Counters:—A Gary-Cummings counter is provided for the main shaft. Gary-Cummings engine log system is installed, with one log on the bridge.

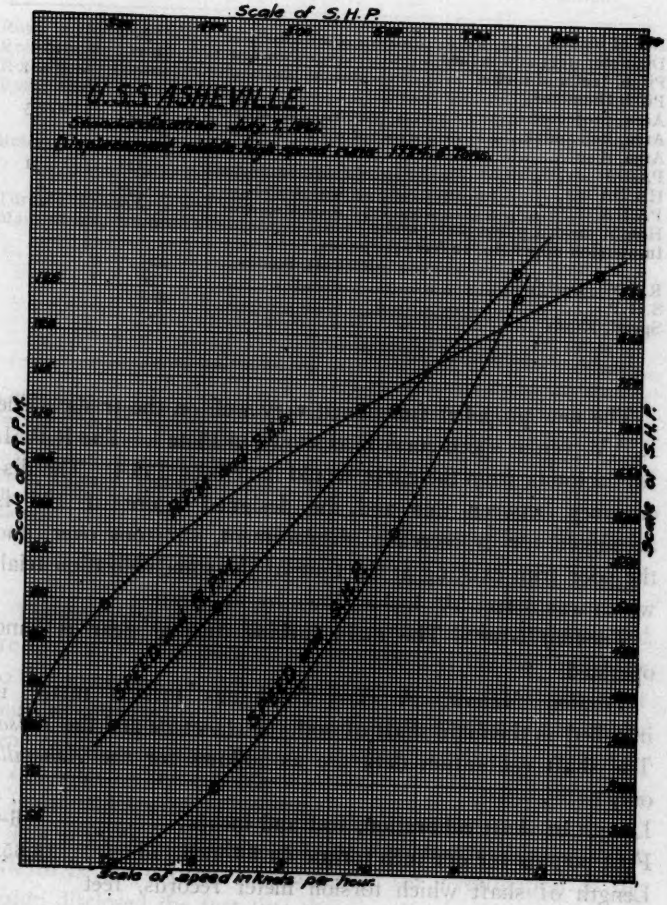

Scale of S.H.P.
U.S.S. ASHEVILLE
Scale of R.P.M.
Scale of S.H.P.
Scale of speed in knots per hour.

PLATE I.

Main Condensing Plant:—At full power, a vacuum of 28½ inches with 60 degrees temperature can be maintained by the main condensing plant which consists of a cylindrical horizontal two-pass condenser, a large vertical twin beam air pump (now used solely as a condensate pump) one Radojet air ejector, and one centrifugal engine driven circulating pump. The condensing plant is located on the port side of the engine room, the condenser being adjacent to and parallel, but above the low pressure turbine and connected to same by a short exhaust trunk built up of steel plate, with a cast piece at the condenser end. The original installation did not contemplate an air ejector which was added because of poor vacuum obtained on dock trial. The air ejector takes suction from the side of the condenser and a baffle is installed inside the condenser to provide for supercooling the air and non-condensable vapors. The dry suction on the condenser to the main air pump was converted into a wet suction by removing the internal stand pipe. Details of parts of the plant are as follows:—

Main Condenser:—

Form—cylindrical (3 feet 6 inches diameter) horizontal.	
Tubes, number	1,102
Tubes, thickness, inch	.065
Tubes, outside diameter, inch	⅝
Tubes, length, feet and inches	6–8⅞
Tubes, ferruled at one end and expanded at other.	
Cooling surface, square feet	1,165
Main exhaust, rectangular, square feet area	4.311
Air pump suction, inches—one 4½ and one 4.	
Auxiliary exhaust connection, inches	8
Safety valve on shells, inches	3–½
Main injection, inches	8
Main overboard discharge, inches	8
Number of passes	2
Feed pump suction direct on condenser, inches	3

Main Air Pump:—

Make	Davidson
Two air cylinders, diameter, inches	14
Two steam cylinders, diameter, inches	7-½
Stroke, inches	10
Suction, inches	6
Discharge, inches	5

The air pump takes suction through a water seal from two connections on the condenser, both wet, and discharges to the filter chamber of the feed tank.

Main Circulating Pump:—

Type: Water balanced centrifugal, engine driven.

Diameter of runner, inches	28
Size of steam engine, inches	5 × 6

This pump can take suction on the sea, engine room bilge or the main drain. It delivers only into the condenser.

Air Ejector:—One, size D-2, Radojet air ejector is installed to remove non-condensible gases. The ejector was manufactured by the C. H. Wheeler Company.

Forced Lubrication System:—The turbine and gearing bearings are supplied by a forced lubrication system having two pumps, each one of which is of ample capacity to serve the bearings at full power of the main propelling machinery; two oil coolers, one having been added after the trials; and necessary piping.

Two inch S. & K. duplex oil strainers are provided in the discharge from the oil pumps. On the *Asheville,* after dock trials, a number of the leads of piping were increased in size, trouble having been experienced in obtaining a sufficient pressure at the bearings to insure positive lubrication. All modifications made to the *Asheville* are being duplicated in the *Tulsa.*

The original lubricating oil pumping plant called for only one pump 3½ × 4 × 4 V.S. which was later changed to two 4½ × 5 × 6 V.S. Worthington pumps. These pumps each take suction from a drain tank of 149 gallons capacity and dis-

charge through the coolers, in parallel, to the turbine and gear bearings and supply oil to the gear teeth through sprayers arranged to supply all pinions for ahead running only. Each bearing and each sprayer has an independent control valve.

Data on the coolers are as follows:

Type	C.S.	No. of Tubes	Dimensions of tubes
No. 1 horizontal Schutte-Koerting No. 2 type Z water through tubes	38.97	87	$\frac{9}{16}\times 39\frac{3}{8}$ ×21 BWG
No. 2 No. 105 Reilly Vert. Multiwhirl.....	12.4	64	$\frac{3}{4}\times .049$

The following oil tanks are provided:

	Capacity gallons each
One drain tank	149
Two settling tanks, fitted with steam coils on *Asheville* only	80
One cylinder oil	5
One auxiliary lubricating oil (omitted on *Tulsa*).....	50
One turbine new oil storage.....	500
One dynamo oil storage.....	75
Four small hand service tanks, total capacty.....	40
Total storage for new oil, *Asheville*.....	670
Total storage for new oil, *Tulsa*.....	620

One DeLaval centrifugal lubricating oil purifier, motor driven, is installed in the engine room. This purifier is size No. 300 capacity 75 gallons/hour, similar to the purifiers provided for destroyers.

Auxiliary Condensing Plant:—There is one Davidson auxiliary condenser for handling all port auxiliary exhaust steam. The condenser is cylindrical, horizontal, two pass, tubes packed and ferruled tubes at one end and expanded at other end, straight tube, set fore and aft and located to starboard of the main turbines.

Data are as follows:

Cooling surface, square feet	300
Number of tubes	305
Diameter of tubes, inch	⅝
Thickness of tubes, inch	.065
Length of tubes as fitted, feet and inches	6–2⅝

The air and circulating pumps are combined and are set on the condenser foundation. The pump has 6-inch steam cylinder 8-inch air cylinder, 8-inch water cylinder and 8-inch stroke. The builder was M. T. Davidson Company.

Feed System:—For feeding the boilers there is provided one main and one auxiliary feed pump, both Davidson vertical simple 8 × 5 × 12.

The main feed pump is located on the forward bulkhead in the engine room and has suction pipes from main feed tank, air pump suction fitting and a manifold connecting to the reserve feed tanks, with discharge pipes to the main feed line passing through the feed heater, with a by-pass, and thence to the main feed check and stop valves on each boiler. A branch leads to the air chamber on the main check valves.

The auxiliary feed pump is located in the fire room. It has suction connections as follows:—independent lead to main feed tank, suction manifold connecting to all reserve feed tanks, deck filling line, and hose connection primarily for pumping down boilers. The discharge connections are as follows:—auxiliary feed line connecting to each boiler through auxiliary feed check and stop valves, hose connection for washing out boilers primarily and a connection to the reserve feed tank manifold. The pump may draw from any reserve feed tank and discharge to any other reserve feed tank. The suction manifold also has a connection to the distilling condenser fresh water pump discharge for filling reserve feed tanks, and a connection to the main condenser for use in taking make-up feed by vacuum.

The main feed tank is located in the starboard forward corner of the engine room. The capacity is 452 gallons. A horizontal partition divides the tank in two parts, the upper part being of about 90 gallons capacity and fitted as a filter tank. Steam drains pass through a coil in the lower tank and discharge to the filter chamber. Overflow and vent pipes are provided.

The total capacity of the reserve tanks is 45.66 tons or 12,282 gallons.

One pressure type feed heater is provided, data being as follows:

	Asheville	*Tulsa*
Make	Schutte-Koerting	Griscom-Russell
Style	Straight tube, 2 pass	Reilly D No. 7, Multcoil
Number of tubes or coils	24	7
Diameter of tubing, inches	⅝	1
Heating surface, sq. ft.	30	43.05
Capacity lbs./hr. from 90 degrees to 240 degrees	18,000	25,500

Flushing Pumps:—The flushing main is supplied by two Worthington centrifugal pumps of capacity 500 gallons per minute at a 60-foot head. The pumps are each driven by Terry steam turbines, 4 H.P. The *Tulsa* is to have only one pump. These pumps replace an original installation of motor driven centrifugal pumps which were removed as they caused an overload on the generating plant.

Fire and Bilge Pumps:—There are two fire and bilge pumps, Davidson vertical simple, size $8 \times 7 \times 12$. Suction connections are sea, secondary drain and hose. Discharge is provided to fire main, overboard, hose connection, distilling condensers and to ice machine condenser.

Steam and Exhaust Piping:—The main and auxiliary steam pipes are of seamless drawn steel and the exhaust piping is copper.

The 5-inch main steam line begins in the after part of the fire room and runs through an air lock to the engine room where after joining the main throttle, it reduces to 4 inches diameter with leads to the H.P. ahead turbine, the astern turbine, and by an emergency portable connection to the L.P. ahead turbine. The main steam line in the fire room is fed by 3½-inch branches from each boiler. The auxiliary steam line which is 2½ inches diameter, takes off the main steam line at its forward end, with a 2½-inch branch to the middle boiler, only, cutting in on boiler side of the main stop. It then supplies, by branch leads, all auxiliary machinery, heating system, galleys, etc., in the fire room and forward of same. It, then, enters the engine room near the center line, runs to starboard, where a 2½-inch branch connects with the main steam line on turbine side of bulkhead stop, thence leads to port and aft, supplying turbo-generators, engine room auxiliaries and machinery aft of engine room including heating system, etc.

A 5-inch auxiliary exhaust line leads from just aft of the middle boiler to the engine room where it passes through a "back pressure" relief valve and enters an 8-inch exhaust pipe connecting the main and auxiliary condensers and which acts as exhaust pipe for the turbo-generators. Branch exhaust pipes from the auxiliary machinery cut into the fore and aft 5-inch line.

Boilers:—There are three "Bureau Express" type small tube water-tube boilers located abreast in one water-tight compartment. The original design called for fitting the boilers for burning either oil or coal. When coal is burned, the two registers are removed and openings blanked. When oil is burned, the two furnace and ash pan doors are removed and the openings blanked. This type of boiler is best suited for oil burning, and the Department, after receiving service reports from the *Asheville*, directed that oil be the primary fuel, increased the oil capacity of both the *Asheville* and *Tulsa* by converting

compartments from coal bunkers to oil tanks, adding 92,000 gallons to the oil capacity, nearly tripling the original capacity, and directed that the *Tulsa* be fitted for burning oil only. When burning coal on the *Asheville* in the tropics, the fire room temperature was unbearable and owing to the poor grade of coal obtainable in Central American and Southern ports, overheating of the uptakes and torching occurred. A small shoulder baffle has been added where the outer rows of tubes join the steam drum, which, it is hoped, will, to some extent, reduce the uptake temperature.

The data for each boiler are as follows:

Designed steam pressure, pounds gage	200
Later increased to	215
Heating surface, square feet	1,316.6
Grate surface, square feet	32.83
Furnace volume, cubic feet	442.25
Steam drum (1), inches diameter	30
Water drums (2), niches diameter	15½
Tubes, diameter 1⅜ inches, number	122
Tubes, diameter, 1⅜ inches, number	946
Down corners (2), diameter inches	6½
Furnace, number	1
Furnace doors, number	2
Burners, number	2
Safety valve, type Twin, size, inches	2½
Forced draft (closed fire room) inches water	2

Smoke Pipe and Uptakes:—The uptakes from the three boilers join in a single smoke pipe which has a height of 70 feet and 9 inches above the boiler grate level. The smoke pipe has an oval section and an area of 15.9 square feet. Dampers were installed originally but have since been removed. The construction of the uptakes and smoke pipe conform to the usual naval practice.

Forced Draft System:—Two forced draft blowers supply the air necessary for combustion at full power, operating on the closed fire room system and taking suction from the port and starboard fire room ventilators by a "Y" branch in the

ventilators just below the main deck. The other branch of the ventilators leads direct to the fire room for use when under natural draft and the port ventilator is rigged with a hoist.

The original design of the vessels called for a single type ZVM Terry turbo blower with semi-silent fan, rated at 8,500 cubic feet of free air per minute at total head of 3½ inches of water and 1,700 R.P.M. This blower was inadequate, and, the necessity for two blowers being apparent after service test of the vessel, two blowers have been installed as follows:—*Asheville:* type ZA Terry turbo blower (similar to Eagle boat installation) rated at 5,000 cubic feet free air per minute at 8 inches of water total head, at 1,860 R.P.M. It is expected that one of these blowers will easily provide the air necessary for combustion at full power, particularly in view of the fact that the Bureau forced draft type of fuel oil registers have been replaced with the Bureau natural-forced draft type which requires relatively much lower air pressures. The fan is a Green 26½-inch semi-silent. The axis of the fan is horizontal. *Tulsa:* type Z Terry turbo blowers with Keith fans (ex Marietta) rated at 56 H.P. at 1,300 R.P.M. The axis of the fan is horizontal.

Fuel Oil Burning System:—A complete fuel oil burning system is installed which consists of the following apparatus:

Two burners per boiler, total of six. Registers, Bureau of Engineering standard natural forced draft type. Atomizers, Bureau standard type using 5/64-inch tips for port use and 3/32-inch tips for sea use. The original design called for Bureau forced draft registers which have recently been replaced by the natural-forced draft type on the *Asheville,* the latter being much more efficient, easier to operate, and requiring lower air pressure. The oil pressure used varies from 150 pounds to 250 pounds, depending on load.

Two fuel oil service pumps, Worthington vertical, duplex size 4½ × 2¾ × 4 inches. The pumps have suction from all oil storage tanks direct or from booster pump discharge, the

latter being used when heavy oil (under 16 degrees Baume) is being burned or when difficulty is experienced in obtaining suction direct from the tanks. The discharge is to the atomizers via the fuel oil heaters.

Two fuel oil booster pumps, Worthington, vertical, duplex size 4½ × 5 × 6 inches. Suction connections are provided to the oil storage tanks direct and the suction manifold is connected to the deck tank filling line. Discharge is provided to all storage tanks in order that oil may be transferred from one tank to another, which may be done without interference with the service of the boilers, to the fuel oil service pump suction and to overboard. All of these fuel oil pumps are located in the boiler room.

Fuel Oil Heaters:—The *Asheville* has 4 vertical cylindrical multicoil type A-19 standard water system heaters, having 6.8 square feet heating surface each, steam inlet 1 inch, steam drain ¾ inch, fuel oil inlet and outlet 1 inch. The *Tulsa* is to be provided with 2 Griscom-Russell Company's Reilly multicoil No. 2 heaters, capacity 2,000 gallons per hour, heating surface 12.3 square feet each.

One fuel oil hand pump, triplex, Rumsey type, size 1⅜ × 3 inches, used for raising steam when there is no steam on the ship.

The fuel oil piping is of steel, lap welded, and flanged. The fuel oil storage tanks have flat vertical heating coils installed around the suction pipes, the coil for each tank having 4 square feet of heating surface. The drains from all fuel oil heating coils are led to an open inspection tank, which drains by gravity to the main feed and filter tank. All storage tanks are provided with low suctions only, except tanks A-8 and C-1 which also have high suctions. These two tanks are intended for service tanks using the high suction normally, the low suction to be used for removing water accumulations. Duplex strainers are provided on suction and discharge of the pump. A fuel oil meter is installed in the service pump discharge.

Refrigerating Plant:—A two-ton refrigerating plant is provided, having ample storage space for meat and chilled fresh provisions.

The *Asheville* has two one-ton, Johns-Manville, dumbbell type, sulphur-dioxide, type No. 4 A-S motor driven machines. The rating is based on 85-degree injection water, and, when outside air is at 100 degrees F., each machine is capable of maintaining 15 degrees in the meat compartments, 35 degrees in the chilled compartments, providing 200 pounds of ice per day and cooling the necessary drinking water from 90 degrees to 50 degrees. One $3\frac{1}{2} \times 4 \times 4$-inch Worthington, vertical, simplex pump provides the SO_2 condenser cooling water. Brine circulation is used, there being two Davidson vertical simplex brine pumps size $4 \times 6 \times 6$ inches.

The refrigerating machines were originally located in the engine room, but have since been moved by the ship's force to the second deck port side between frames 57 and 64. A 200-pound-per-day ice making box is provided as well as a brine cooled scuttle butt.

The installation on the *Tulsa* is a duplicate of that on the *Asheville* except that two one-ton Kroeschell carbonic anhydride steam driven machines are provided. They are installed in the compartment originally intended as a wet provision store room on the platform deck between frames 59 and 62. The *Tulsa* machines have 2×4-inch compressors and the steam engine is $5\frac{1}{2} \times 7$ inches. The compressors displace 3,500 cubic inches at 150 R.P.M.

Distilling Plant:—The distilling plant has a rating of 5,000 gallons of 2 grain per gallon water when operated in single effect. The *Asheville* has 2 Bureau, vertical shell, "U" tube evaporators, having $36\frac{1}{8}$-inch tubes, heating surface 50.75 square feet each. Two 3,000-gallon Bureau distilling condensers are provided, each having $162\frac{5}{8}$-inch tubes, 3 feet and $1\frac{5}{8}$ inches long, cooling surface 39.8 square feet each.

There is one evaporator fed heater, Bureau type, of 16.49 square feet heating surface.

The plant is designed for high pressure operation in single or double effect. Provision is made for use of orifice method of operation and use of continuous blow.

The pumps provided for the distilling plant are: one feed pump, Davidson V.S. $3\frac{1}{2} \times 4 \times 4$; one distilling condenser fresh water pump, of ditto size and type. Distilling condenser circulating water is taken from the flushing main.

The distilling plant on the *Tulsa* is similar to that on the *Asheville* except as follows: 2 No. 6 Reilly submerged, multi-coil evaporators are provided. The evaporator coils will be replaced by "Lillie" type tube nests, using the film method of evaporation. Larger distilling condensers and air and brine pumps will be provided. The plant will be operated in double effect only on the low pressure-vacuum system, and it is expected that the average daily output extending over 3,000 hours' continuous operation will easily exceed 5,000 gallons. This plant will be a great improvement over that installed on the *Asheville* in regard to economy, ease of operation and comfort to the operating personnel.

Air Compressing Plant:—For cleaning boilers, and miscellaneous use in machine shop and elsewhere there is a Westinghouse $11 \times 11 \times 12$ steam driven air compressor.

Electrical Plant:—There are two General Electric 35 K.W. turbo-generators; turbine type G-A, 3,400 R.P.M.; generators class H, 125 V., 280 Amperes; and one main switch board located in the engine room starboard. The voltage is 125.

The lighting system is two wire feeder system, 125 volts, and has about 450 regular fixtures, and 35 12-inch portable bracket fans.

There are two 24-inch H.P. searchlights and one 12-inch incandescent signal searchlights.

Signal and running lights are as follows:

Standard port and starboard blinker set.

Portable tube blinker.

Complete set of running lights and anchor lights.

Absence, man-o'war, man overboard and break down lights.

Truck, steering and wake lights.

A complete interior communication set is installed consisting of the following circuits:

	Voltage
Officers' calls	20
Voice tube calls	20
General alarm-gong system	125
Shaft Revolution indicator	125
Rudder indicator system	125
Submarine signal receiving system	125
Salvo firing system	125
Whistle operator	125

The 20-volt systems obtain current through a resistance mounted on the main switchboard, the voltage being reduced from the ship's operating voltage, 125.

The list of motors is as follows:

Motors for	H.P.
No. 1 Ice machine (*Asheville* only)	3
No. 2 Ice machine (*Asheville* only)	3
Machine shop, shaper	2
Machine shop, grinder	1
Machine shop, lathe	7½
Machine shop, drill press	2
Ventilating	22
Galley	5
Laundry	4
Fresh water pump	6

The radio installation consists of the following:

One 2 K.W. arc transmitter for long range work with power supplied by one 3 K.W. motor generator.

One 2 K.W. spark transmitter for medium range work, with two 3 K.W. motor generators.

One model BC motor buzzer transmitter for short range work.

One medium range receiver for 250 to 7,000 meter wave lengths.

One long range receiver for 1,000 to 25,000 meter wave lengths.

One Radio telephone set type CW 936 for short range work.

One field radio set, 1/4 K.W., with power supplied by gas engine.

The main antenna is flat top 85 feet long, 4 wires spaced 3 feet 9 inches apart. The lead is 75 feet long.

The telephone antenna is 2 wire, vertical type, spacing 2 feet, length 60 feet.

Shop Equipment:—One general work shop is provided which is excellently well equipped for a vessel of this size. The machine tools are all motor driven and are as follows:

One extension gap lathe 19 inches × 38 inches × 7 feet centers, driven by 7½ H.P. motor.

One 15-inch shaper, with 2 H.P. motor.

One 28-inch upright drill press, 2 H.P. motor.

One grinder, wet and dry wheels, 1 H.P. motor.

A complete set of blacksmiths' and coppersmiths' hand tools and a portable forge are also furnished.

Trials:—As is customary with vessels built at Navy Yards, only dock trials were held prior to commissioning of the vessel, on 17th July, 1920. After commissioning, but before leaving the Navy Yard for service, a short sea trial was held with representatives of the Navy Yard present. The vessel then saw service in Central American waters and, upon returning to the United States in May, 1921, she was docked at the Navy Yard, Charleston, apparatus installed for measuring water and oil, and a full set of standardization trials, 4-hour

full power trial and a 20-hour 10-knot endurance trial were conducted July 7-9, 1921, off Provincetown, Mass., under the supervision of the Board of Inspection and Survey. All trials were successfully carried out.

The standardization was held first. The data obtained is plotted on Plate I. The maximum speed attained was 11.52 knots (.48 knot below expected speed) on 127.26 R.P.M. and 847 S.H.P. The estimated displacement at the middle of the high speed runs was 1,724.5 tons or 149.5 tons over normal displacement of 1,575 tons. The trim was 3 feet 4 inches by the stern. These two factors account for failure to obtain 12 knots. During the standardization the weather was overcast and hazy, wind on beam, force 1-2, sea smooth.

On 8th July the steering gear was tested. With the vessel going ahead full speed the rudder was thrown full right and, after completing a circle, thrown full left. The time to move the rudder from full right to full left was 15 seconds; the estimated angle of heel when turning was 3 degrees outward and the turning circle was 500 yards in diameter. The ship makes a complete circle in four minutes.

The next test was to throw the engines from full speed astern from full speed ahead. The vessel was dead in the water in 3 minutes 55 seconds and the head reach was 660 yards. The engines were slow in backing but since raising the astern steam chest pressure from 50 to 75 pounds a much better performance is obtained.

The steering gear was then tested with the engines making 90 R.P.M. astern. The rudder was moved from full left to full right in 16 seconds.

The next test was to throw the engines from full speed astern to full speed ahead when making 94 R.P.M. astern. The vessel was dead in the water in 1 minute and 5 seconds and stern reach was 120 yards. The engines were at 123.5 R.P.M. ahead in 2 minutes and 25 seconds.

The hand steering gear was next tested, the rudder being put from 25 degrees right to 20 degrees left in one minute. The time to change from steam to hand steering and vice versa was excessive, being nearly five minutes.

The anchor gear was then tested by dropping the anchor in 25 fathoms of water, veering to the bitter-end (120 fathoms) and then heaving in at full speed which was at the rate of 9 fathoms per minute.

Upon the completion of the above tests the 4-hour full speed trial and 20-hour 10-knot endurance run were held, data being obtained as follows. On the trials the fuel used was coal. Readings on the full power trial were taken every 15 minutes and on 10-knot run every half hour.

	4 hour Full Speed	20-hour 10-knot
Speed, knots	11.48	10.19
R.P.M. of propeller	126.69	111.79
Slip of propeller, per cent	14.59	14.08
S.H.P., main turbines	838	577
Displacement middle of trial, tons	1673.5	1626.5
Trim by stern, beginning, inches	40	...
Total coal consumption, pounds	10167.04	47701.0
Average coal consumption, per hour, pounds	2541.76	2385.05
Average coal consumption for all purposes per S.H.P. of main turbines only, pounds	3.033	4.134
Average coal consumption per knot at average speed, pounds	221.4	234.1
B. T. U. per pound coal	14469	14469
Total water evaporated, pounds	92021.00	400948.0
Water evaporated per hour, pounds	23005.25	20047.4
Water evaporated for all purposes per S.H.P. of main turbines, per hour, pounds	27.453	34.744
Evaporation (actual) per pound coal, pounds	9.051	8.405
Make up feed used during trials, gallons	592	2754
Pressures:		
Main steam at boilers, pounds gauge	201.4	193.4
Main steam at engines, pounds gauge	198.3	192.3
H. P. turbine steam chest, pounds abs	165.9	129.1
L. P. turbine steam chest, pounds abs	15.3	15.2
Main condenser, vacuum, inches Hg	27.7	27.3
Auxiliary condenser, vacuum, inches Hg	20.3	Not used
Barometer, inches Hg	...	...
Auxiliary exhaust, pounds gauge	10.0	10.0
Excess auxiliary exhaust to	aux. cond.	m. cond.
Square feet main condensers per S.H.P.	1.390	2.019

	4-hour Full Speed	20-hour 10-knot
Temperatures, degrees, Fahrenheit:		
Main injection	67	65.7
Main overboard discharge	136	136.5
Main steam	360	343.1
Outside air	65.0	65.0
Engine room, floor plates	118.8	120.0
Fire room, floor plates	120.7	113.2
Air pump discharge	...	...
Feed tank	...	...
Feed to heater	139.4	153.1
Feed from heater	197.4	204.1
Lubricating oil to cooler	120.5	116.3
Lubricating oil from cooler	103.4	99.9
Boilers in use, number	3	3
Boilers, sq. ft. H. S. in use	3950	3950
Sq. ft. H. S. per S.H.P.	14.141	20.537
Water evaporated (actual) square feet H.S./hour	5.82	5.08
Coal burned per square foot G. S./hour	25.8	24.7
Air pressure in fire room, inches water	.14	.36
Forced draft blower (1) R. P. M.	1700.	2154.8
Main air pump double strokes/minute	19.4	24.5
Main circulating pump double strokes/minute	250.5	213.6
Main feed pump double strokes/minute	26.3	22.3
Auxiliary feed pump double strokes/minute	20.8	30.5
Fire and bilge pump double strokes/minute	25.1	16.8
Lubricating oil pump double strokes/minute	44.9	42.5
Dynamo, number in use	1	1
Dynamo, volts	123.2	115.9
Dynamo, ampères	259.6	249.0
Steam steering engine in use	Yes	Yes
Steam on galley	Yes	Yes
Fog horn sounded every minute, hours	2	20
Lubricating oil cooler circulating water from	Fire Main	Fire Main
Steam air ejector in use	Yes	Yes
Sea, state of	Mod. gr. swell	Choppy
Courses	Various	Various
Wind force and direction	SW. 2	SW. 2-3

ELECTRIC DRIVE ON BATTLESHIPS.

By Alexander Mark Charlton, Lieut. Comdr., U.S.N., Member.

INTRODUCTION.

The application of the induction motor for the propulsion of ships presents such a radical departure from the time honored reciprocating engine and the later development of the turbine, that most naval officers are bewildered by the apparent complexity of the machinery layout in an electric drive ship. The literature on the subject of electric propulsion is meagre at best and is usually so technical and presupposes such a knowledge of alternating current phenomena, that the average person finds himself again and again compelled to go back to fundamentals. This usually results in an abandonment of the whole subject as one too complex for easy assimilation and it remains as much a mystery as ever.

It is believed that a simple explanation of the systems of electric drive now installed on the battleship in commission, together with a discussion of the principles of the induction motor, will be of interest to all officers in the service. It is the hope of the writer that in the following pages such a simple exposition of the subject of electric propulsion will be found.

GENERAL.

The use of electric motors for the propulsion of ships is not a new conception, as patents were taken out on this type of propulsion as early as 1893.

In 1908 two fireboats for the city of Chicago were equipped for electric propulsion by the General Electric Company. Turbine driven generators furnished power for the propulsion motors which were mounted directly on the propeller shafts.

The General Electric Company was very enthusiastic on the subject of electric propulsion, especially for capital ships and submitted plans for such a system for both the *Wyoming* and the *Pennsylvania* when they were authorized. The Navy Department was not prepared at either time for such a departure, but when three colliers of the same displacement were authorized, the opportunity for electric propulsion was at hand. These colliers were the *Cyclops,* fitted with reciprocating engines, the *Neptune,* fitted with turbine reduction gear, and the *Jupiter,* fitted with electric motors.

The performance of the *Jupiter* set at rest all fears as to the reliability of electric machinery for ship propulsion. At the same time the economy of her installation over that of her two sister ships was quite marked.

The increasing size of battleships and higher speed demand, called for a radically improved method of propulsion. Reciprocating engines had virtually reached the apex of their development and direct turbine drive was admittedly imperfect because the turbine is inherently a high speed machine, while an efficient propeller must run at low speed. In combining them directly the efficiency of each was sacrificed to run at an intermediate speed.

Our Navy Department, assured of equal reliability, increased economy, and practically the same weight for electric propulsion machinery as compared with reduction gear drive, found the military advantages of the former so preponderant, that it definitely committed itself to the policy of electric drive for capital ships. This decision was made at the time the *Idaho, Mississippi* and the *New Mexico* were being designed, and the last named was selected for the first battleship installation. As will be seen later, full advantage could

not be taken of the flexibility of arrangement possible with electric drive machinery on account of the advanced hull plans and it was designed to fit into spaces arranged for turbine machinery. The contract for the *New Mexico's* machinery was made with the General Electric Company.

In the next two battleships authorized, the *Tennessee* and the *California,* which were designed for electric propulsion from their inception, full advantage of the possibilities of this system was taken. The Westinghouse Electric and Manufacturing Company was given the contract for the machinery of the *Tennessee* while that for the *California's* apparatus was let to the General Electric Company.

The remaining ships authorized to date—ten battleships and six battle cruisers, were all designed for electric propulsion; six battleships and two battle cruisers to have Westinghouse machinery and four battleships and four battle cruisers to have General Electric equipment.

To decide on such a machinery program before the final trials of the first electric drive battleship were completed required an implicit faith on the part of the Navy Department in this type of apparatus, but its faith has been fully justified by the results obtained on the ships in commission.

The advantages of electric propulsion for capital ships have been well summed up by Rear Admiral C. W. Dyson, U. S. Navy, in an article entitled "Development of Machinery in the U. S. Navy During the Past Ten Years." (JOUR. A. S. N. E., May, 1917):

"1. Greatly increased torpedo protection for ships;
2. Greater flexibility in machinery arrangement;
3. Better and wider separation of important units;
4. Minimum lengths and diameters of steam pipes;
5. Fewer bulkheads pierced by steam and feed piping;
6. Reduced heating of vessel from steam pipes;
7. Better centralization of power;

8. Reduced engine-room complement;

9. Elimination of danger from fractures of piping due to shells striking protective deck;

10. Greater ease in control;

11. Greater flexibility in power distributed;

12. Better maintenance of economy through a wide range of powers;

13. No metallic contact between rotor and stator of motor;

14. Elimination of all dangers of disarrangement due to shaft vibration when the helm is put hard over;

15. Maximum reduction in length of shafting;

16. Increased backing power.

In opposition to these claims can only be advanced one, that of greater weight. This can be discounted on the heavy ships where protection is of vital importance."

How these advantages are utilized will appear in the following pages.

THE EQUIPMENT.

Electric drive is installed at present on the *New Mexico*, *California* and *Maryland* (General Electric type), and on the *Tennessee* (Westinghouse type). The *Colorado* (building) is to have Westinghouse machinery, and the *West Virginia* (also building) is to be equipped with General Electric apparatus. The remaining ships of the 1916 program are, at present writing, to be scrapped. The machinery as designed for the above battleships may be divided into the following constituent parts:—

1. A high speed turbine, direct connected to an alternating current generator which generates polyphase current;

2. A direct current generator driven by its own turbine, which furnishes excitation for the alternator;

3. Induction motors (rotors mounted on the propeller shafts) which receive their energy from the alternator;

4. Control apparatus for the motors and turbines;

5. Auxiliaries for the main and exciter turbines and for the boilers.

Of this apparatus, the turbo-alternator, its exciter and the auxiliaries are in the engine room, the motors are in the motor rooms, while the control apparatus is in a separate compartment called the control room (*Tennessee* et seq.). The *New Mexico* has the control apparatus in the center engine room where the auxiliaries are concentrated.

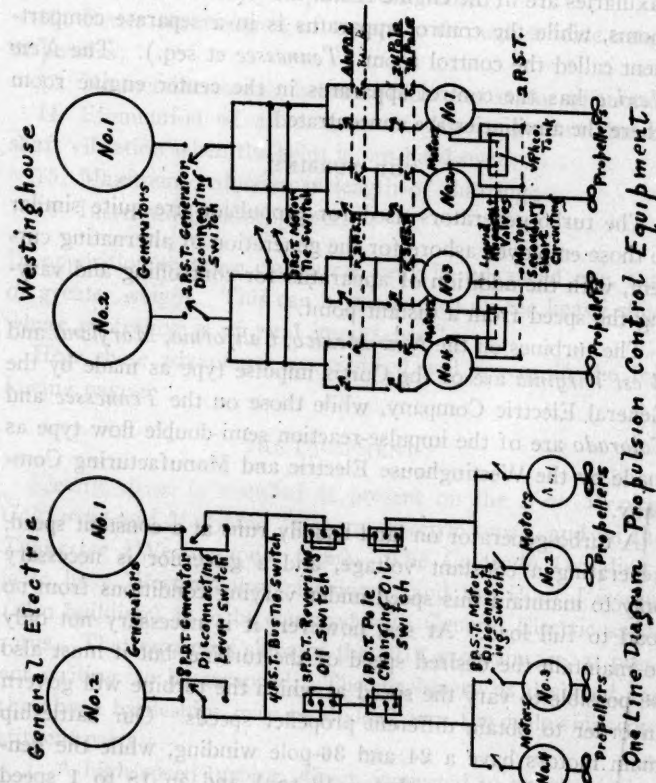
THE TURBINE.

The turbo-generators used for propulsion are quite similar to those employed ashore for the generation of alternating current, with the addition of apparatus for controlling and varying the speed from a distant point.

The turbines on the *New Mexico, California, Maryland* and *West Virginia* are of the Curtis impulse type as made by the General Electric Company, while those on the *Tennessee* and *Colorado* are of the impulse-reaction semi-double flow type as made by the Westinghouse Electric and Manufacturing Company.

A turbo-generator on land usually runs at a constant speed, generating a constant voltage, and a governor is necessary only to maintain this speed under varying conditions from no load to full load. At sea, however, it is necessary not only to maintain the desired speed of the turbine, but it must also be possible to vary the speed at which the turbine will govern in order to obtain different propeller speeds. Our battleship main motors have a 24 and 36-pole winding, while the generators are 2-pole, so that a 12 to 1 and an 18 to 1 speed reduction is possible. At full speed of the turbines these ratios correspond to approximately 21 knots and 15 knots.

In order to obtain other speeds and to vary the speed by small increments necessary for keeping station in formation a special type of governor is necessary.


Westinghouse
No.1
No.2
Generators
2 R.S.T. Generator Disconnecting Switch
2 R.S.T.
No.1
No.2
No.3
No.4
Propellers
General Electric
No.1
No.2
Generators
3 R.D.T. Generator Disconnecting Lever Switch
3 R.S.T. Bus Tie Switch
3 R.D.T. Reversing Oil Switch
3 D.P.S. Pole Changing Oil Switch
3 R.S.T. Motor Disconnecting Switches
Motors
Propellers
One Line Diagram Propulsion Control Equipment

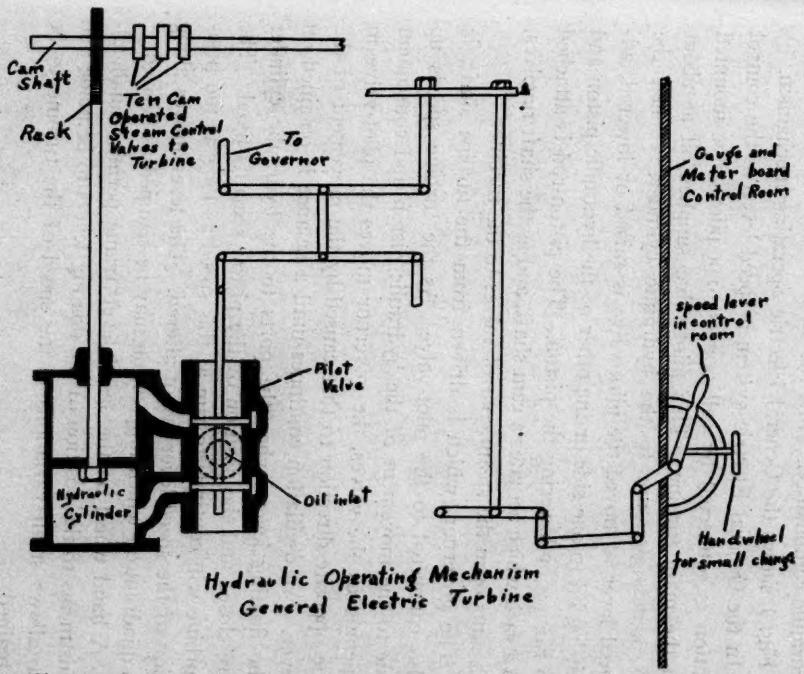


FIG. 1.—HYDRAULIC OPERATING MECHANISM GENERAL ELECTRIC TURBINE.

The methods of governor control on the General Electric and Westinghouse turbines are quite different but apparently equally satisfactory. The first named system is a mechanical-hydraulic combination, while the latter is entirely hydraulic.

Fig. 1 shows the General Electric operating mechanism.

In the General Electric system a speed lever in the control station is connected by bell cranks to a pilot-valve mounted on the main turbine. This pilot-valve supplies oil at about 75 pounds pressure to the hydraulic cylinder. When the speed lever is moved the pilot-valve is raised or lowered, admitting oil to one side or the other of the hydraulic piston and so raising or lowering the piston. The piston-rod is attached to a rack which actuates a cam-shaft, and as the shaft revolves, its cams open the steam control valves to the turbine.

The governor, which is driven from the turbine shaft, is also connected to the pilot-valve As the turbine speeds up due to the movement of the hydraulic piston and consequent opening of the valves, the governor moves the pilot-valve in the opposite direction to that caused by the movement of the lever. An oscillation continues until a balance is obtained on the floating levers when the ports to the hydraulic cylinder are closed and the position of the piston remains fixed. The turbine will continue to run at this speed as long as the position of the speed lever is not altered. The lever moves along a quadrant on which its position may be clamped as desired.

A hand wheel on the speed lever frame permits small adjustments to the position of the floating levers to be made and so allows small variations in the speed of the turbine to be obtained.

The Westinghouse type of speed control is entirely hydraulic. A speed wheel in the control room operates, through a worm and bell crank system, a governor control valve. This valve has an oil pressure of about 80 pounds from the lubricating system up to it. The movement of this valve by the

speed wheel allows any oil pressure from zero to the maximum to be sent to the hydraulic governor on the turbine. This governor is in principle a dead weight fly-ball governor in which variable speed is obtained by varying the dead weight. To do this, the effect of gravity in the ordinary construction is replaced by the pressure from a hydraulic piston against which the variable pressure from the governor control valve is sent. Thus if the pressure is 5 pounds per square inch, the centrifugal force due to the turbine speed of about 400 R.P.M. will balance the piston, but if the pressure is increased to 60 pounds, full speed of the turbine (2100 R.P.M.) will be necessary to balance it. The hydraulic piston and the fly-ball of the governor are connected by levers to the stem of the governor relay valve. Movement of this valve admits oil to one side or the other of the piston in the operating cylinder. The movement of this piston controls the movement of the steam inlet valve which admits steam to the nozzle chamber of the turbine. The amount of steam actually getting into the nozzles is determined by hand operated valves at the turbines.

The system is thus seen to consist of two parts: (*a*) The hydraulic system from the control room, which in effect changes the dead-weight governor so that it will govern at the speed desired, and (*b*) the governor system itself.

Fig. 2 shows the Westinghouse governor system.

The action of the governor is as follows: When a pressure is put on the hydraulic piston from the control room, point A moves to the left. The floating levers moving about the fixed point B pull down the governor relay valve and oil is admitted to the under side of the piston in the operating cylinder, and the piston moves up. The system now operating with C as a fulcrum, opens the governor controlled inlet valve and steam is admitted to the turbine which speeds up. At the same time B is raised which moves the governor relay valve up and stops the flow of oil to or from the operating cylinder. As it speeds up, the fly-balls on the governor go out, pulling

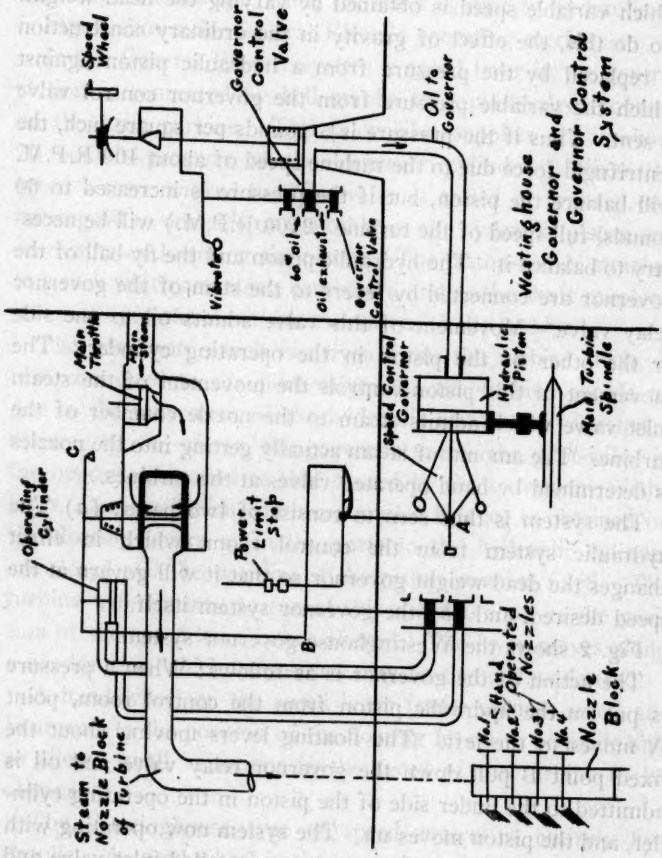

Speed Wheel
Governor Control Valve
Oil to Governor
Vibrator
To oil
Oil exhaust
Governor Control Valve
Westinghouse Governor and Governor Control System
Main Throttle
Speed Control Governor
Hydraulic Piston
Main Turbine Spindle
Operating Cylinder
Power Limit Stop
Steam to Nozzle Block of Turbine
No. 1 Hand Operated Nozzles
No. 2
No. 3
No. 4
Nozzle Block

FIG. 2.

A to the right and tending to pull the governor relay valve up further and thus allow the oil pressure access to the top of the operating cylinder. A balance soon obtains and the turbine continues to turn at a constant speed as long as the position of the speed lever and hence the oil pressure on the hydraulic piston remain constant.

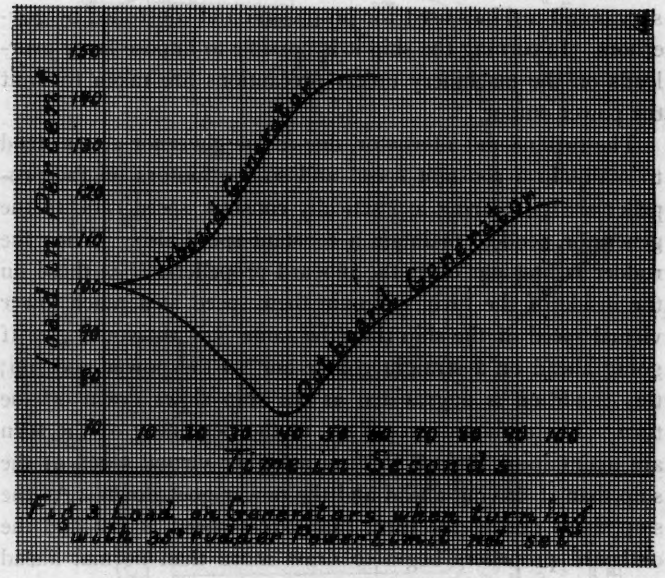
In connection with the steam control to the turbine it is necessary now to describe the "power limit" device which is necessary on all turbines used for propulsion and controlled by governors. In turning the ship it is possible to impose large overloads on the machinery unless precautions are taken to limit the power output.

The turbine and alternator, due to the governor action, tend to maintain a constant speed and so the motors on the propeller shafts, bearing a practically constant speed ratio to the generator, tend to maintain a constant speed also. When the rudder is first put over, the inboard propellers, *i. e.*, those on the inside of the turn, maintain their speed at a much lower virtual speed of the ship than the same R.P.M. would give if going ahead on a straight course and so the power goes up; this condition is aggravated as the ship slows down on the turn. The outboard propellers, at the beginning of the turn are maintaining their R.P.M. at a higher virtual speed of the ship than if driving ahead and the power goes down. As the ship slows during the turn, this excess speed is lost and the outboard propellers begin to draw more than normal ahead power.

Figure 3 shows the manner in which the inboard and outboard sides vary when making a turn with 35 degrees rudder. The curve is from tests of a battleship making 19 knots and turning with 35 degrees rudder. In this test the turbines were allowed to take any load which might be imposed up to the maximum capacity of the machine. The power of the generator supplying the inboard side increased rapidly during the first 50 seconds at which point it reached the maximum

output of the turbine. The power of the generator supplying the outboard side falls to about 70 per cent at the end of 40 seconds and then increases to about 118 per cent at the end of 100 seconds.

It is evident that if precautions are not taken to prevent such overloads being imposed on the machinery it would be necessary to carry sufficient excitation on the generators con-



tinuously to take care of such overloads. At reduced loads this would present no difficulty except lack of economy. At maximum loads, however, an unwarranted reserve capacity would be necessary in the generator fields thus resulting in larger generators than would be needed ordinarily. To overcome this condition, a power limit device is provided to limit the amount of steam the turbine can take to a predetermined value.

This device on both the General Electric and Westinghouse turbines consists of a mechanical stop on the floating lever system between the governor and the inlet valves which may be set at any desired point, and so limit the amount of steam going to the turbine. In the General Electric system the power limit stop is set by means of a set of levers operated from the control room, while in the Westinghouse system, it is set by means of a small motor and worm gear mounted on the turbine and operated from the control room. In both systems signal lights are fitted to show the position of the power limit stop. The power limit device is arranged in both types so that it will not interfere with the governor in slowing down the turbine.

In the General Electric turbine the entire flow of steam is regulated by the governor control, there being ten inlet valves operated in succession by the valve cam shaft as power is demanded. The Westinghouse turbine has a number of hand operated nozzles on the turbine which are opened as necessary for various ranges of speeds. The governor controls the flow of steam through each nozzle up to the limit of that nozzle. Thus on the *Tennessee,* number 4 nozzle gives a maximum speed of ten knots, number 3 a maximum speed of fifteen knots, numbers 2 and 3 (on both turbines) a speed of twenty-one knots, the design being arranged to operate up to the full opening of one nozzle before opening another.

Both types of turbine are fitted with emergency governors which will function when the maximum designed speed of the turbine is exceeded, due to failure of the main governor when the load is removed. The emergency also functions when the oil supply fails. This device closes the main throttle when it operates. Hand trips, which will also close the main throttle, are fitted to be operated from the control room and at the turbine.

In both types of turbine the governor controls the admission of auxiliary exhaust steam to the lower stages of the turbine.

Gland sealing in the General Electric type is by steam alone. Leakage from the high pressure end of the turbine passes through a labyrinth packing and thence to the low pressure end of the turbine where it again passes through a labyrinth packing and thence to the atmosphere. When starting up and when running at low speed, live steam is supplied to the high pressure packing. At high power, where the leakage from the high pressure packing is in excess of the requirements of the lower pressure packing, the excess steam is delivered to the ninth stage of the turbine through an unloading valve.

Diaphragm packing sleeves are provided on all diaphragms to prevent steam leakage due to the pressure differences between the stages.

The Westinghouse turbine is double flow on the low pressure portion, that is, a low pressure section is on each end of the turbine. To prevent the entrance of air into the turbine along the shaft, a combined steam and water seal is provided at each end of the turbine. When standing by, or at any speed below half of full turbine speed, a labyrinth type of gland is sealed with steam, but upon reaching about 1000 revolutions, a speed governor in connection with a gland control valve shuts off the steam and admits water to the outer portion of the gland which is fitted with a centrifugal pump runner. The water from the boiler main feed line, reduced to a pressure of about 7 pounds, is admitted to the runner chamber at the periphery, submerging the runner on each side to such a depth that the centrifugal pumping head produced by the runner balances the total water pressure. Thus, as both sides of the runner are covered by water to a certain depth, no air can pass into the exhaust chamber of the turbine.

The main bearings on both types of turbine are supplied with forced lubrication from the lubricating pumps. There

are four of these pumps on each of the ships, all of the Kinney rotating plunger type. These pumps also supply oil for the governor systems and the main motor bearings.

GENERATORS AND MOTORS.

The conditions under which the generators and motors for electric propulsion operate are so different from those under which similar apparatus ashore may operate that it is well to consider them together.

Ashore, the generator is usually situated in a central plant and the voltage must be maintained at a constant potential. Various uses are made of the current generated which is distributed, sometimes over very long transmission lines, for lighting systems as well as power. Most of the induction motors connected run at constant speed and constant load or if not, the speed is varied at the motor. The generator runs at a constant speed and the direct current field is varied only as necessary to keep the voltage constant. The exciter voltage is varied by cutting resistance in or out of the exciter field automatically, in order to change the field current.

Aboard ship the problem is entirely different. The main generators are used solely for propulsion, the only connected load being that of the motors on the propeller shafts. Since the motors must run at various speeds in order to get the desired revolutions of the propellers and must run under various conditions of load due to backing and turning of the ship, and state of the wind and sea, the speed of the generator and the field current must be capable of variation by the operator at will. The variation in speed of the generator is accomplished by changing the speed of the turbine as previously described. The method of field control and its necessity will be taken up in detail later.

The generators of the General Electric and Westinghouse types are very similar in appearance and construction. Each

consists of a stator (armature) and a rotor (rotating field) with housing, end shields, collector, brush holders, ventilating apparatus, etc.

The stator of each type consists of a cylindrical steel framework, on the inner surface of which is placed the stator winding. This consists of a number of well insulated conductors laid side by side in the laminated iron core all around the frame and suitably joined by connectors at the ends to form the type of winding desired.

The General Electric stator is wound two-phase and the windings are so grouped that by manipulating an 8-pole double-throw switch between the generator and motors, a voltage of 3,000 or of 4,242 may be obtained from the generator. This is accomplished by making the generator winding in four parts and bringing both ends of each part out to a terminal board. The connections are so made that throwing the 8-pole switch in one direction gives the square, one circuit, high voltage arrangement, and throwing it the opposite way gives the diametrical, 2-circuit, low voltage arrangement. Figure 4 shows the method of varying the connections.

The Westinghouse stator is wound three-phase and generates 3,270 volts at full speed.

The ventilation of both types of generator is accomplished by fans on the rotor which draw air in from under each end of the stator, push it through the air gap and discharge it through the stator core to a trunk leading to the deck. Dampers are provided so that the generator may be closed up when idle.

In both the General Electric and Westinghouse generators the rotor is a solid one-piece forging slotted to receive the coils. The rotors are two-pole and receive their current from the exciter generator through brushes bearing on collector rings.

As the exciting current is fed in to the rotor from the exciter and the rotor is revolved by the turbine, an electro-motive

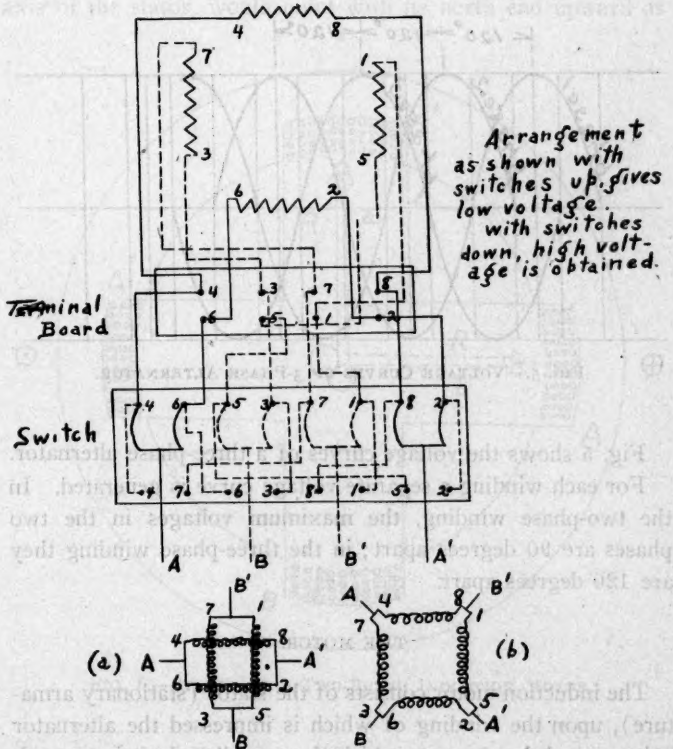


FIG. 4.—SWITCHING ARRANGEMENT FOR HIGH AND LOW VOLTAGE. GENERAL ELECTRIC ALTERNATORS.

force is generated in the armature of the alternator. The shape of this voltage curve depends on the winding of the armature.

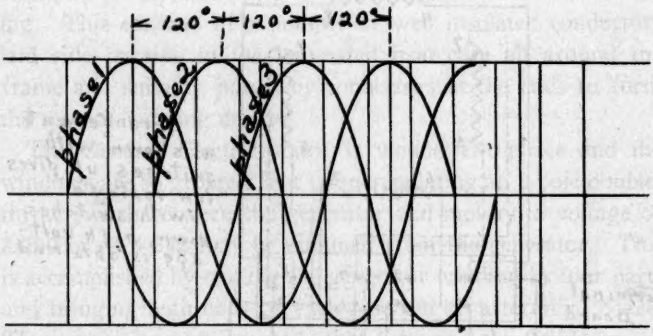


FIG. 5.—VOLTAGE CURVES OF 3-PHASE ALTERNATOR.

Fig. 5 shows the voltage curves of a three-phase alternator.

For each winding a separate voltage curve is generated. In the two-phase winding, the maximum voltages in the two phases are 90 degrees apart; in the three-phase winding they are 120 degrees apart.

THE MOTOR.

The induction motor consists of the stator (stationary armature), upon the winding of which is impressed the alternator voltage, and the rotor, to which the propeller shaft is secured.

The production of the torque necessary to turn the rotor of the motor and thence the propeller, depends upon a movement of the flux or polar regions on the stator around the circumference of the rotor. This fact is very important and quite difficult to follow and it will be gone into in detail.

Figure 6 represents a stator of a two-pole two-phase induction motor. When current flows positively through phase A, that is from A toward A′ through the rear connectors, while no current flows in phase B, a compass-needle, pivoted at the axis of the stator, would point with its north end upward as

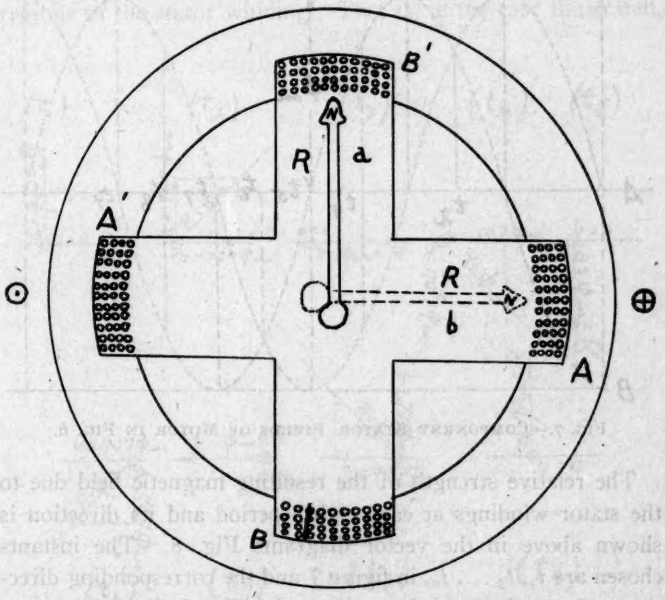


FIG. 6.—STATOR OF A TWO-PHASE INDUCTION MOTOR.

shown by the full arrow. When current flows positively in phase B, that is from B to B′, while no current flows in A, the north end of the needle would point to the right as shown by the dotted arrow. Suppose now that the phases A and B are connected to a two-phase generator so that currents flow in each of them, equal in value and having 90 electrical degrees phase difference.

Figure 7 shows the values and directions of the component stator fields of the motor at the various instants during one and one-half cycles. The field due to phase A leads that due to phase B by 90 degrees.

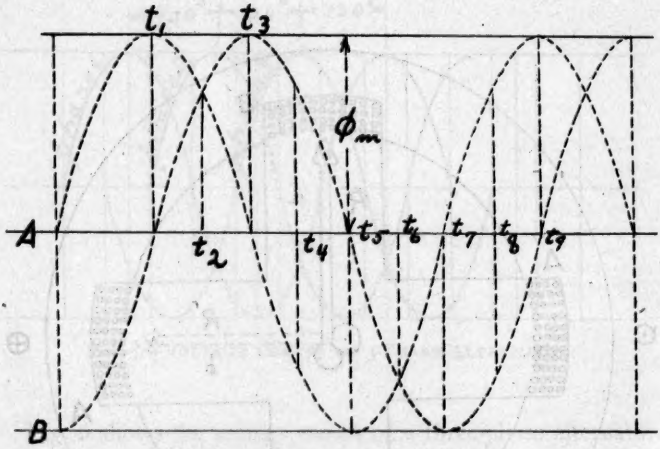


FIG. 7.—COMPONENT STATOR FIELDS OF MOTOR IN FIG. 6.

The relative strength of the resulting magnetic field due to the stator windings at each eighth period and its direction is shown above in the vector diagrams Fig. 8. The instants chosen are t_1, t_2 . . . t_9, in figure 7 and the corresponding direction and magnitude of the resultant field R is indicated by t_1, t_2, t_3 . . . . t_9 in Fig. 8. At t_1 the current in phase A has its maximum positive value and the current in phase B is zero. Consequently the resultant field R is the same as that due to phase A alone. Thus if φ_m represents the maximum strength of field or amount of flux due to maximum current in either phase alone, R is equal to φ_m and is directed vertically upward at instant t_1. At t_2, 1/8 period later, the field due to phase A has been reduced to the value $a = \varphi_m$ cos. 45 degrees $= .707\varphi_m$, although still positive and upward; at the same time the field

due to phase B has increased from zero to the value $b = \varphi \sin 45 \text{ degrees} = .707\varphi_m$ and its direction is positive and to the right. The vector sum of these components $= \sqrt{2} \times .707 \varphi_m = \varphi_m$. If we follow the field during a complete cycle from t_1 *to* t_9 we will find that any given magnetic polar region on the stator will travel over the distance occupied by two polar regions of the stator winding. That is, in the case illustrated,

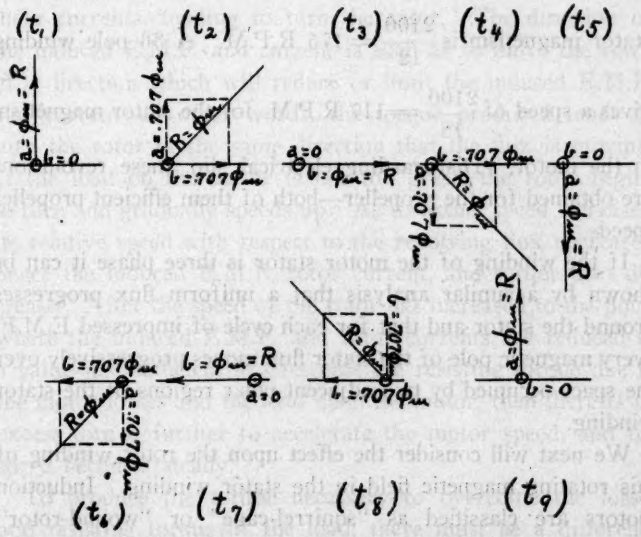


FIG. 8.—VECTOR DIAGRAM OF MAGNETIC FIELD OF MOTOR, FIG. 6.

although there are always two magnetic polar regions on the surface of the stator (diametrically opposite, it being a two-pole winding), both of these poles will sweep around the axis while keeping a fixed relation to each other, making one complete revolution in the stator for each cycle of E.M.F. impressed on the stator windings, which means 60 revolutions per second if the frequency is 60 cycles. By a similar analysis,

it can be shown that if the stator were wound for four poles instead of two, the time of two cycles of impressed E.M.F. would be required for each of the four poles to move completely around the stator, or the synchronous speed of the stator magnetism would be 1/2 of 60, or 30 revolutions per second, corresponding to a frequency of 60 cycles in the generator. This fact is taken advantage of in obtaining low propeller speed. With a two-pole generator making 2,100 R.P.M. and a 24-pole stator winding on the motor the speed of the stator magnetism is $\frac{2100}{12} = 175$ R.P.M. A 36 pole winding gives a speed of $\frac{2100}{18} = 117$ R.P.M. for the stator magnetism in the motor. Disregarding electrical slip, these revolutions are obtained for the propeller—both of them efficient propeller speeds.

If the winding of the motor stator is three phase it can be shown by a similar analysis that a uniform flux progresses around the stator and that for each cycle of impressed E.M.F. every magnetic pole of the stator flux moves progressively over the space occupied by two adjacent polar regions of the stator winding.

We next will consider the effect upon the rotor winding of this rotating magnetic field in the stator winding. Induction motors are classified as "squirrel-cage" or "wound-rotor" motors according to the method of placing in the rotor slots the copper conductors in which the stator magnetism induces the currents which react to produce the torque of the motor. A squirrel-cage rotor winding consists of a number of copper bars in the rotor slots secured to a ring at each end so that all conductors are electrically connected. A wound-rotor winding is very similar to that of a stator, there being a definite winding for each phase. These windings may be brought out to a rheostat which allows us to control the speed of the motor,

to improve the power factor or to adjust the starting torque. The latter possibility only is taken advantage of in our navy propulsion motors.

Consider now the rotating flux in a squirrel cage stator. When the rotor is standing still, this moving flux induces E.M.F. and currents lengthwise of the rotor from the points where the flux is most dense toward the points where the flux is less dense or of the opposite sign. The rotating flux exerts a mechanical force upon the conductors which carry these currents, tending to turn the rotor. The direction of the induced E.M.F. and current is such as to move the rotor in a direction which will reduce or limit the induced E.M.F. and current; in other words, the torque produced tends to turn the rotor in the same direction that the flux is moving. If the load on the motor is not too great, the rotor begins to turn and gradually speeds up. As its actual speed increases, its relative speed with respect to the revolving flux decreases, hence the induced E.M.F. rotor current, and torque also decrease. After the speed of the rotor has increased to the point where the induced E.M.F. and rotor currents are reduced to a value just sufficient to overcome the resisting torque due to the motor losses and the load upon the motor, then there is no excess torque further to accelerate the motor speed, and the speed becomes steady.

To produce the torque necessary to overcome the losses and resisting torque of the load, there must be a difference in speed between the rotor and revolving stator field, else there would be no induced E.M.F. nor current and so no torque. This difference in speed, the difference between the actual and synchronous speed is called the "slip" and is usually expressed in per cent of the synchronous speed.

The torque developed by the rotor is proportional to the product of the field strength and the rotor current.

To all intents and purposes the rotor circuit may be looked upon as one of resistance and inductive reactance in series, and

supplied with a varying E.M.F. The electromotive force results from the rotation of a magnetic field of nearly constant value (for any one given set-up in a naval motor), its speed varying in respect to the rotor according as the rotor goes fast or slowly. If the rotor is stationary, the magnetic field in the stator is moving past the rotor at 100 per cent speed. In case of sudden reversal of the motors while the ship is going ahead this magnetic field in the stator may be moving past the rotor at 160 or 170 per cent (propellers going ahead and field reversed). The torque characteristics under these conditions are peculiar and interesting. Before proceeding with the description of the motor torque characteristics let us first examine the propeller torque characteristics. Fig. 9 shows the propeller torque curves when reversing the motors with the ship going ahead full speed. When power is cut off the motors, the speed of the propellers immediately drops to about 75 per cent of the full ahead speed. If now the motor connections are reversed, there is a torque opposing that caused by the motion of the ship on the propellers. This propeller torque rises to nearly 100 per cent of full load torque as the speed of the propellers is reduced to 40 per cent of their full speed revolutions. The propeller torque then falls off to about 35 per cent of full load torque while the propellers are at standstill. As the propellers are reversed the propeller torque rises again until at 100 per cent torque, the motors are making 40 per cent of the full ahead speed in the astern direction. (This curve is based on the assumption that the speed of the ship through the water does not change during the reversing operation. As it does slow down, the rotational speed obtained when backing with full load torque is higher than the 40 per cent noted above.)

Ordinarily in going ahead, when the rotor is allowed to turn freely under the action of the torque developed, it attains such a speed that the magnetic field is moving past the rotor at 1 per cent or less of its 100 per cent speed. That is, the rotor

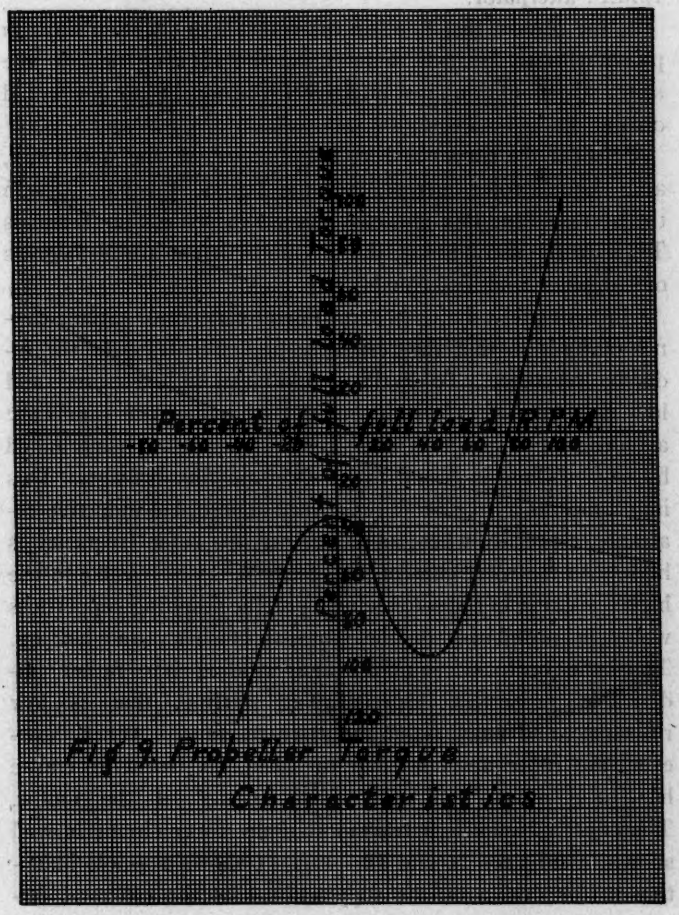


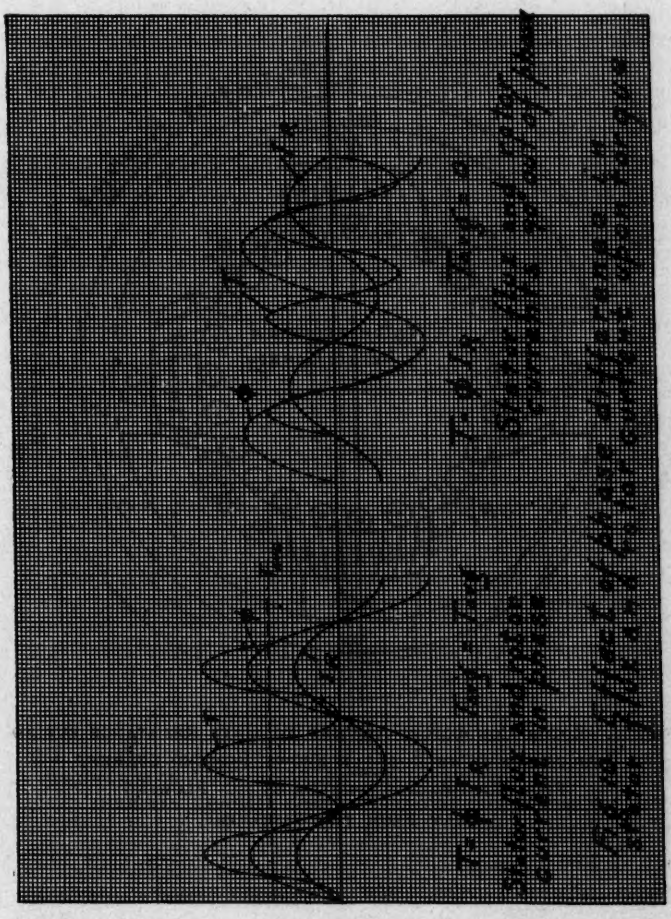
Fig 3. Propeller Torque
Characteristics

circuit may be looked upon as connected to a constant field, variable speed, and hence, a variable frequency and variable E.M.F. alternator.

If the current that flows in the rotor circuit were always in phase with the E.M.F. induced by the rotating field in the stator, the condition most effective for producing torque would obtain.

Consider the curves in Fig. 10. If φ equals the stator flux, and I equals rotor current, then the Torque T equals $I\varphi$. In the first case the flux and currents are in phase, and T equals $T_{avg.}$ In the second case, the flux and current are 90 degrees out of phase and T_{avg} equals zero.

Unfortunately the ordinary squirrel cage rotor circuit cannot be built so that the resistance predominates over the inductive reactance. To do this would mean high resistance and low inductance. A certain amount of inductance cannot be avoided; the conductors must be in slots, that is surrounded by iron. There is bound to be a certain flux leakage. This inductance as a rule produces a very appreciable inductive reactance, especially when, as at starting, the rotor frequency is high. As the rotor currents are heavy it is not feasible to use high resistance in the rotor winding as the resulting copper loss would be prohibitive on the score of efficiency and heating. Thus the ordinary squirrel cage winding has rather undesirable properties, especially at starting, that is low resistance, and unavoidable inductive reactance. When, however, the rotor has come up to speed, the rotor reactance has little effect upon the characteristics of the motor. The effect of inductive reactance in the rotor circuit is well shown by Figures 11 and 12. Fig. 11 shows a 4-pole induction motor drawn on the supposition that the electrical circuits are non-inductive—that they offer resistance but have zero inductive reactance. Then at any instant the conductors which have the greatest induced E.M.F. have also the greatest current flowing through them, and this current flows in the



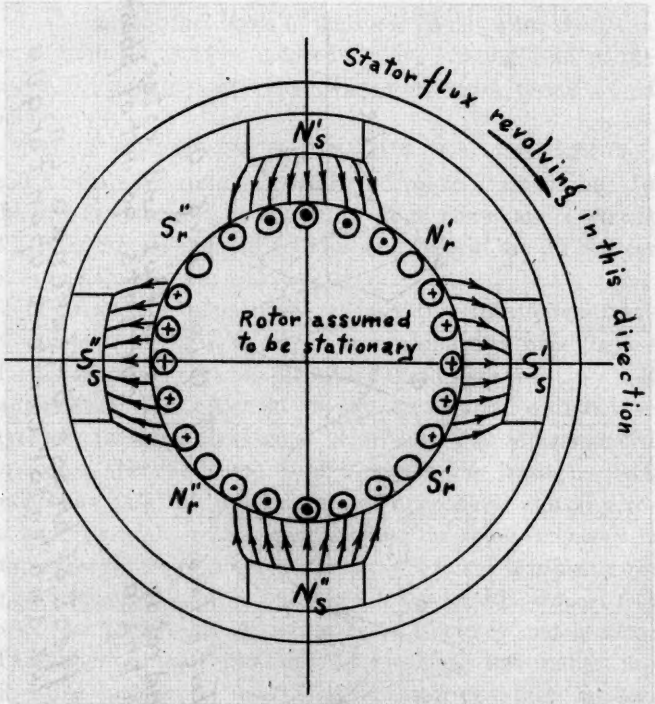


FIG. 11.—ROTOR CONDUCTORS ASSUMED TO HAVE RESISTANCE BUT NO INDUCTANCE. THE DOT ⊙ SHOWS INDUCED CURRENT FLOWING TOWARDS, THE CROSS ⊕ FLOWING AWAY FROM THE OBSERVER.

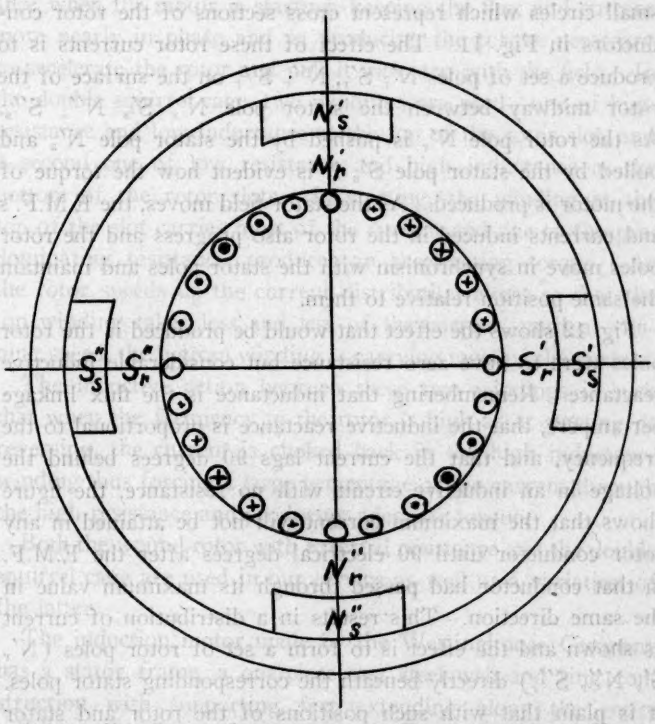


FIG. 12.—THE ROTOR CONDUCTORS ARE HERE ASSUMED TO HAVE REACTANCE BUT NO RESISTANCE. POLES ON THE ROTOR ARE DIRECTLY OPPOSITE SIMILAR POLES ON THE STATOR.

direction of the induced E.M.F. The relative magnitudes of these currents in the various rotor conductors is indicated roughly by the relative size of the direction symbols in the small circles which represent cross sections of the rotor conductors in Fig. 11. The effect of these rotor currents is to produce a set of poles N'_r S'_r, N''_r S''_r on the surface of the rotor midway between the stator pole N'_s S'_s N''_s S''_s. As the rotor pole N'_r is pushed by the stator pole N'_s and pulled by the stator pole S'_s it is evident how the torque of the motor is produced. As the stator field moves, the E.M.F.'s and currents induced in the rotor also progress and the rotor poles move in synchronism with the stator poles and maintain the same position relative to them.

Fig. 12 shows the effect that would be produced if the rotor poles were to have zero resistance but considerable inductive reactance. Remembering that inductance is the flux linkage per ampère, that the inductive reactance is proportional to the frequency, and that the current lags 90 degrees behind the voltage in an inductive circuit with no resistance, the figure shows that the maximum current will not be attained in any rotor conductor until 90 electrical degrees after the E.M.F. in that conductor had passed through its maximum value in the same direction. This results in a distribution of current as shown and the effect is to form a set of rotor poles (N'_r S'_r N''_r S''_r) directly beneath the corresponding stator poles. It is plain that with such positions of the rotor and stator poles no torque can be produced.

The above discussion shows the necessity for high resistance in comparison with the inductance when starting an induction motor under load, and in the case of a ship, when suddenly reversing the direction of rotation of the motors. To avoid these poor starting qualities, the rotor can be wound like the stator and the terminals of the winding connected through slip rings to external resistance to be used for starting purposes only, or a double squirrel cage rotor may be employed

which avoids the disadvantage of slip rings necessary to connect the windings to an external resistance. In the first method, the external resistance predominates over the inductive reactance when the motor is starting, keeping the flux and current more nearly in phase and so producing the torque necessary to accelerate the rotor and pull it into step with the field. In the double squirrel cage, two windings are used—one of high resistance and low inductance at the top of the rotor slot, and a second one of low resistance and high inductance at the bottom of the rotor slots. In starting, the winding at the top of the slot carries most of the current and due to the predominating resistance, produces an accelerating torque. As the rotor speeds up the current distribution alters so that the top winding takes less and less of the current, until at running speed, the bottom winding is carrying most of the current.

The inductive action between these two windings is such that when the frequency in the rotor is high as at starting or reversing, the current is choked back in the high inductance winding thus forcing a large percentage of the current through the high resistance and producing adequate torque.

Both the wound rotor with external resistance and the double squirrel cage are used in our service as well as a variation of the latter.

The induction motor made by the Westinghouse Company has a stator frame, a one-piece cast steel web and ring construction with supporting feet extending along the entire length and cast integral with the frame. A groove is planed in the under side of each foot and there is a corresponding key on the foundation track in the ship. This arrangement is provided for sliding the stator forward when access is desired to the internal parts of the motor. An extension shaft is put in the hollow end of the rotor shaft and carried to a bearing on the bulkhead of the motor room. The rotor is not disconnected from the propeller shaft but is held in place while the stator is moved forward along the track far enough to uncover the rotor.

The stator laminations are dovetailed into ribs in the frame casting and are clamped between heavy end-plates with fingers for supporting the teeth.

Longitudinal ventilating ducts run the whole length of the stator through the laminations, and a gap at the center of the stator is left in the laminations so that air may be drawn from the rotor.

Two separate three-phase windings are laid in the stator slots, a 24-pole and a 36-pole winding. This arrangement was considered preferable to a single winding on account of the complicated system of connections necessary with the latter, and also because with a single stator winding connected for 24 and 36 poles, it would be impracticable to cut out individual coils from the stator winding, a generally accepted and quick method of putting the motor back in service in case of local breakdown. Each slot contains two coil sides of each winding, the 24-pole coils being located at the bottom of the slot.

The windings in the slots are insulated with mica, paper and shellac, while the curved ends of the windings are insulated with treated cloth repeatedly varnished. The whole winding, after assembly and connection, is given six coats of varnish, and the whole stator baked after each coat. The coils are held in the slots by bakelite micarta wedges driven into grooves at the top of the teeth.

The rotor is built up on a strong double arm spider, made of cast steel in one piece; it is securely pressed on to and keyed to the shaft. The laminations are dovetailed into the spider and clamped between heavy end plates with fingers for supporting the teeth. The rotor laminations have ventilation holes the entire length of the rotor with a gap in the laminations opposite the gap in the stator laminations.

The rotor is wound with a single three-phase winding, permanently connected to three collector-rings. This winding is adapted for either 24 or 36-pole operation by a special connection.

Fig. 13 represents a section equal to one-sixth of one phase of the rotor winding. The coils are arranged in groups, and by means of group connectors A, are connected for 24 poles in two parallel circuits joined in "star" as indicated at V. With the stator connected for 24 poles, the motor will operate in the usual manner for induction motors having phase wound rotors permitting suitable external resistance for starting or reversing to be inserted in the rotor circuit.

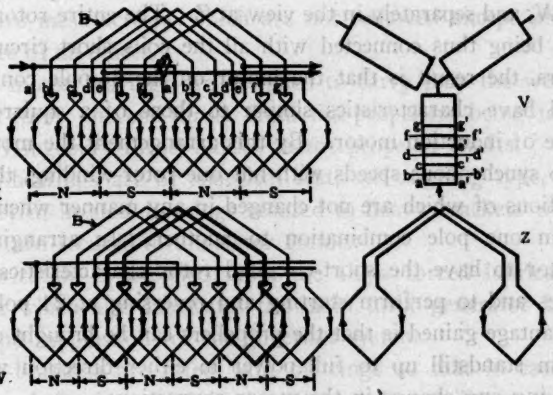


FIG. 13.—SCHEMATIC WIRING DIAGRAM OF MAIN MOTOR ROTOR. U. S. S. "TENNESSEE."

There being two parallel circuits, it follows that certain points of the two circuits will have the same potential. The groups of coils are arranged in such a manner as to locate these equi-potential points at a and a', b and b', c and c', etc., and these points are joined together in pairs by special connectors B. When the motor is operated on 24 poles, these connectors do not carry any load current since they join together points having the same potential.

When, however, the 36-pole stator winding is energized, the conditions change as shown in the view at W, represent-

ing the same section of the rotor windings as at X, so that in the space occupied by four poles in the view at X, for the 24-pole connection, the stator winding, when connected for 36 poles, will produce six poles as indicated by arrows at W, which indicate the direction of the E.M.F.'s of the several coils. The special connectors B now serve as short circuits connecting pairs of coils together in series, with their E.M.F.'s added.

Such a pair of coils is indicated by heavy lines in the view at W, and separately in the view at Z. The entire rotor winding being thus connected with all the coils short circuited in pairs, the result is that the motor on the 36-pole connection will have characteristics similar to those of a squirrel cage type of induction motor. By this arrangement the motor has two synchronous speeds with but one rotor-winding, the connections of which are not changed in any manner when going from one pole combination to another. In arranging the motor to have the short-circuited rotor characteristics at 36 poles and to perform starting and reversing at 24 poles, the advantage gained is that the propellers can be brought quickly from standstill up to full power in either direction without making any change in the motor connections.

The ventilation of the motor is by fans under normal conditions of operation. Two adjustable-speed motor-driven exhaust fans are mounted on top of a sheet-steel casing which encloses the stator frame and serves to confine the cooling air. The cooling air is taken from the main deck through trunks down to the motor room, drawn through wire mesh screens at each end of the motor and thence through the axial ducts in the stator and rotor cores to the radial duct in the center of the cores. This duct leads into the stator housing whence the air is drawn by the fans and discharged up exhaust trunks to the main deck. This is a very effective method of ventilation as the cooling air is brought into close contact with the parts of the machine where the heat is generated, and the paths

of the heat flow are short. In addition to the forced draft fans the rotors themselves are provided with fans so that the motors can operate at full power for short periods of time in case of failure of the exhaust fans.

The longitudinal ducts in the core and the large central radial duct, however, render it very easy for foreign particles to get into the air gaps if they once get inside the rotor, and great care must be taken to prevent such entry into the rotor. The openings at the ends of the rotor are covered with very fine wire mesh screens which are not removed except in the presence of an officer.

The motor bearings are designed to carry only the weight of the rotor, the end-thrust from the propellers being taken up by Kingsbury thrust-blocks on the propeller shafts.

The bearings are of spherical seat, self-aligning type, and are carried in steel housings secured to heavy brackets which are in turn secured to the stator frame.

Lubrication is supplied from the ships forced lubrication system which also lubricates the main turbo-generator bearings and supplies oil for the governor system.

The bearings are fitted with oil rings so that in case of failure of the forced lubrication, the bearings will operate indefinitely at full power without overheating.

In the lower half of the stator a series of direct current electric heaters is fitted for use when the motors are idle for any length of time. The purpose of the heaters is to keep the motor temperature slightly higher than that of the surrounding air and so prevent condensation of moisture with its consequent damage to insulation.

The external resistance for use in starting and bringing the motors into step on the 24-pole winding is provided by liquid rheostats situated in the control room. One rheostat is provided for each of the four motors, but the rheostats are mounted in pairs.

Fig. 14 shows a cross section of one of the double rheostats. It consists of a sheet steel tank, two sets of electrodes, a cooling system, and two motor driven centrifugal pumps. The tank has two main sections, a lower storage tank and an upper section, divided into two electrode chambers and an overflow chamber, the bottom of which opens into the storage tank.

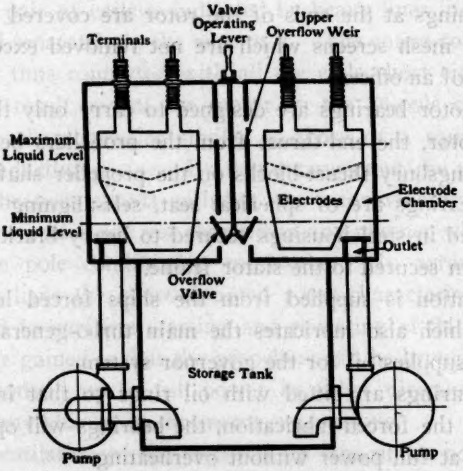


FIG. 14.—CROSS SECTION OF DOUBLE LIQUID RHEOSTAT. U. S. S. "TENNESSEE."

The normal position of the rheostat is with the storage tank full, the electrode chambers filled to the "minimum liquid level" and the overflow valve open.

In this position the maximum resistance is in series with the rotor windings, as the steel electrode plates are almost entirely uncovered. The electrolyte is pumped continuously from the bottom of the storage tank to the electrode chambers whence it returns to the storage tank through the open overflow valve.

This condition exists when the motors are first started. Immediately after starting, the rheostat lever is moved, thereby closing the overflow valve. The liquid now rises in the electrode chambers covering more and more of the electrodes and cutting out more and more resistance until the maximum liquid level is reached. At this point the liquid overflows into the overflow chamber and returns to the storage tank. The resistance is now all cut out and a short-circuiting switch, called the motor secondary, is closed, thus short-circuiting the motor secondary's slip-rings. The rheostat lever is now moved back, opening the overflow valve, and the liquid falls to the minimum liquid level to be in readiness for another operation.

The electrolyte is a 1 per cent solution of sodium carbonate.

With normal condition of the pumps, about fifteen seconds are required for the liquid to rise, and about twenty seconds for it to fall.

The stator frames of the *New Mexico* motors, made by the General Electric Company, consist of cast steel rings of I-beam section suitably spaced and rigidly held in position, being riveted to solid steel castings which form the feet. These feet are bolted to the foundations which form part of the ship structure. The ring sections are held together by boiler plates rolled to shape, and riveted to the rings. The whole frame is so proportioned that the laminations and windings are supported with practically no deviation from a perfect circle. The small air gap necessary in induction motors requires a rigid stator structure so that there will be no chance of rubbing. The boiler plates enclose the structure and direct the forced currents of air by which ventilation is secured. Holes in the upper half of the boiler plates allow the air to pass to the exhaust trunks which extend up from the motors and which support the exhaust fans above the motors. The core is built up of segmental laminations stamped from steel which is enameled to prevent eddy currents. Each segment is independently held by slots to dove-tailed ribs secured to the I-beam sections.

Radial air ducts are provided at frequent intervals along the length of the core by omitting the laminations for a certain distance. These ducts line up with similar ducts in the rotor. There are no longitudinal ventilating ducts as in the *Tennessee's* motors.

There is but a single stator winding of wound coils insulated with mica tape and an external layer of tape with several treatments of moisture-resisting varnish.

In order to get the speed ratio of 2 :3 as obtained by the 24 and 36 pole windings of the motor stators on the *Tennessee* a system of end-connections and switches was worked out which gives 24 and 36 poles with a single winding.

Fig. 15 shows the principle of design, winding connections and pole changing switch.

The upper row of arrows shows the distribution of coils and the direction of current flowing in the two phases of the winding when the motor is 24-pole connected. The second row of arrows shows the corresponding distribution for the 36-pole connection. The third group of arrows shows how the coils are permanently grouped inside the motor into eight winding sections which may be connected by proper switching to make up 24 or 36-pole combinations.

The diagram below shows how these various sections are combined into the motor winding.

The lower diagram shows how the winding sections are arranged with reference to the pole changing switches.

It will be seen from the diagrams that in changing from the 24 to the 36 pole combination some of the winding sections are changed to a different phase with the instantaneous direction of current retained or reversed, and that other winding sections are kept in the same phase with the direction of current retained or reversed.

In making up the design for the *New Mexico's* motors, it was decided, (inasmuch as a resistance inserted in the rotor

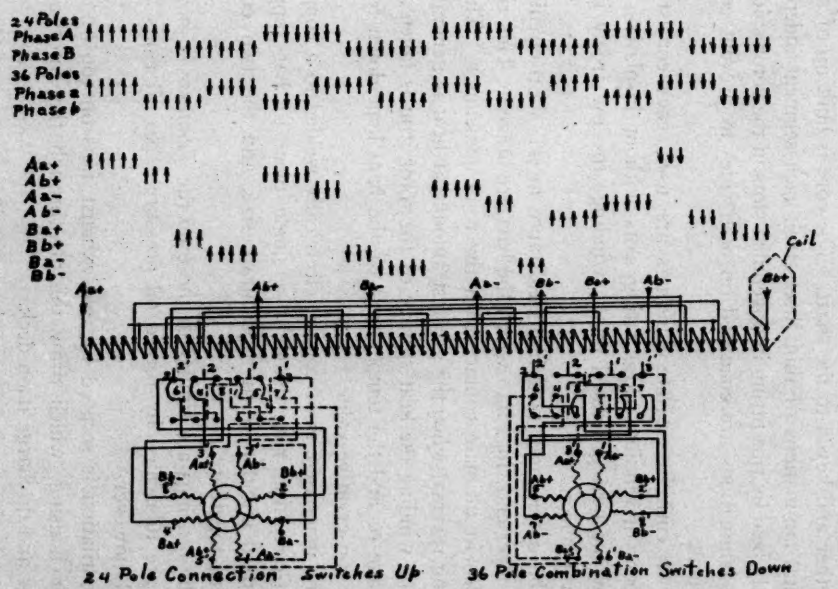


FIG. 15.—DIAGRAM OF 24-36 POLE MOTOR WINDING U. S. S. "NEW MEXICO."

winding would be needed for only a few seconds in starting and reversing,) to adopt a double squirrel-cage, or Boucherot type of motor, thus eliminating the rheostat.

The rotor is built up of two cast steel spider sections bolted together and keyed to the shaft. The core is built up of segmental laminations of thin sheet steel, each segment independently held by ribs fitting in dovetailed slots in the rotor spider. The laminations are enameled on both sides to prevent eddy currents.

Two entirely separate windings are used, each consisting of bars driven into close fitting slots without insulation, to allow the heat generated in the windings to flow freely into the rotor iron.

The outer cage is of high resistance metal, short-circuited by copper end rings into which the bars are brazed. This ring is also in intimate contact with the rotor punchings. Thus, during reversal when the greatest amount of heat is generated in this winding the heat storage of the rotor iron is taken advantage of and the temperature of the bars kept down to a very conservative value.

The inner cage is of copper, electrically welded to a copper short-circuiting ring made of a single piece of bar copper rolled in circular form to accurate size, and the two ends welded together.

The bearings are similar to those on the *Tennessee's* motor and are lubricated by forced oil pressure. No oil rings are fitted, however.

Ventilation is secured by two exhaust fans on top of the stator housing, which draw the air through the rotor and stator and discharge it on deck.

To examine the interior of the motor it is necessary to break the coupling between the motor rotor and the propeller shaft, take the weight of the rotor by slings, loosen the stator end plates, and swing the rotor clear. This is a more difficult

operation than that of opening up a motor on the *Tennessee* where the stator is moved forward.

The motors for the *California, Maryland* and *West Virginia,* also of General Electric design, are similar in appearance and construction to those of the *New Mexico,* except as to the winding of the rotors.

There are two independent windings in the same slots. The winding nearest the air gap is of low resistance, phase wound, and insulated with mica. With the 36-pole arrangement on the stator, the rotor currents have a two-phase distribution. Each phase is connected to a pair of slip-rings on the rotor shaft, one pair at each end of the rotor. The brush leads from each pair of collector rings are connected to a contactor mounted on the end shield, by means of which the winding can be short-circuited or opened from the control room.

At the bottom of the slots are high resistance bars of a squirrel-cage winding. These bars are separated from the phase winding by thick insulating strips and are connected together by end rings.

In starting and reversing, the contactors are left open so that no currents flow in the phase winding. The squirrel cage is thus the active winding, furnishing the torque for acceleration of the motor.

When the motors are in step, the contactors on the phase winding are closed, and this winding carries practically all the current.

The 36-pole combination with the contactors open is always used for starting and reversing. When the contactors are closed with the 36 pole connected, and also when the stator is 24 pole connected, whether the contactors are open or closed, the squirrel cage is relieved of practically all current. When the stator is 24 pole connected, the rotor winding is so arranged that no current passes through the collector rings or brushes.

CONTROL APPARATUS.

For each speed at which the propeller motors are turning over, a certain field current is necessary in the main alternators in order that sufficient current may be induced in the motor rotors to keep the motors in step.

We have seen heretofore that if the alternator voltage, which with constant speed on the alternator is dependent on the exciting current, is allowed to fall off beyond a certain point, the torque is no longer sufficient to keep the rotor at approximately synchronous speed, and it slows down and stops.

The field current for any one speed may vary slightly from time to time, as the load varies, that is, depending on the state of wind and sea, and the position of the rudder. It is necessary, then, to be able to vary the exciter voltage and hence, the field current.

On all of the battleships having electric propulsion, 300 K.W. non-condensing turbo-generator exciters are installed, one for each main alternator. These machines are intended primarily for main drive, supplying the excitation current, as well as that necessary to run a number of electrically driven auxiliaries connected with the main drive. As these auxiliaries must run on constant voltage, and a variable voltage is desired for excitation, a "booster" motor-generator set is introduced between the exciter and the alternator field. The "booster" is a motor-generator the motor of which is driven at constant speed from the exciter whose voltage is normally maintained at 240. The exciting voltage to produce the necessary field current varies through wide limits—the maximum being when the motors are reversed with the ship going ahead at highest speed, and the minimum being when the ship is going ahead at slowest speed. To obtain these wide variations, the generator of the booster motor generator is connected in series with the field circuit of the main alternator. If current is sent

in one direction through the field of the booster motor generator, its voltage is added to that of the exciter voltage and it is said to "boost" the latter; if the current is sent through the field of the booster generator in the opposite direction, it opposes the exciter voltage and is said to "buck" it. The control of this booster field current and hence the control of the main alternator field current is effected by means of a rheostat. The rheostat is controlled by the operator in the control room, the lever being mounted beside the steam lever. Fig. 16 shows an elementary diagram of the rheostat and connections.

The two operations, control of the steam and control of the field, are, of course, the most important in obtaining speed and accuracy of answering "bells" from the bridge. The realization of this led to the design of the control room or cell where are centralized all the means of control and switching for the two main turbo-alternators and the four propulsion motors.

The arrangement and appearance of the control apparatus for the General Electric and Westinghouse equipments are somewhat different, the difference being principally in the appearance of the switches.

The *New Mexico,* with General Electric equipment, was so far along in design when it was decided to install electric propulsion that full advantage could not be taken of the possibilities inherent in this type of propulsion in arranging the machinery.

Her layout is shown in Fig. 17.

There is no separate control room, a portion of the center engine room being designated as a control cell, which contains the main drive breakers. One side of this cell is made up of a gauge and meter board, and the operating levers are grouped in its vicinity.

The *Tennessee* and *Colorado* (Westinghouse), and the *California, Maryland* and *West Virginia* (General Electric) were

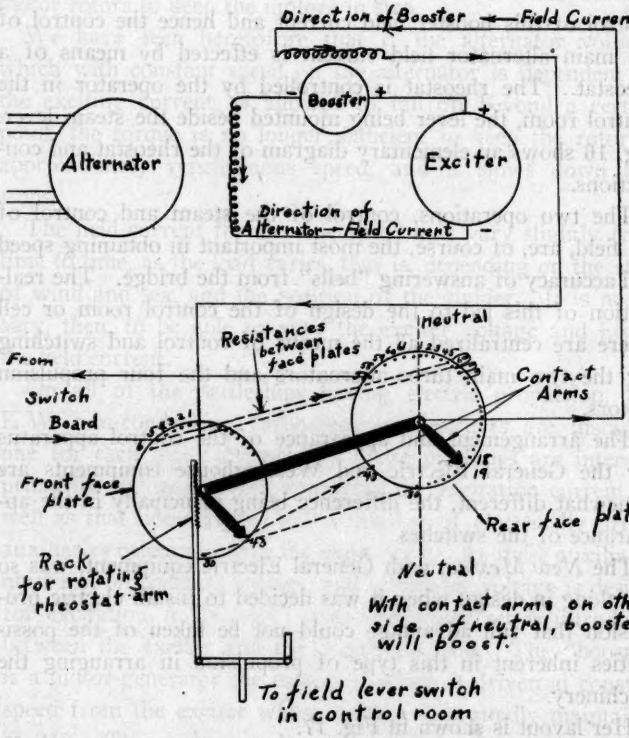


FIG. 16.—BOOSTER FIELD RHEOSTAT. SHOWN IN "BUCKING" POSITION.

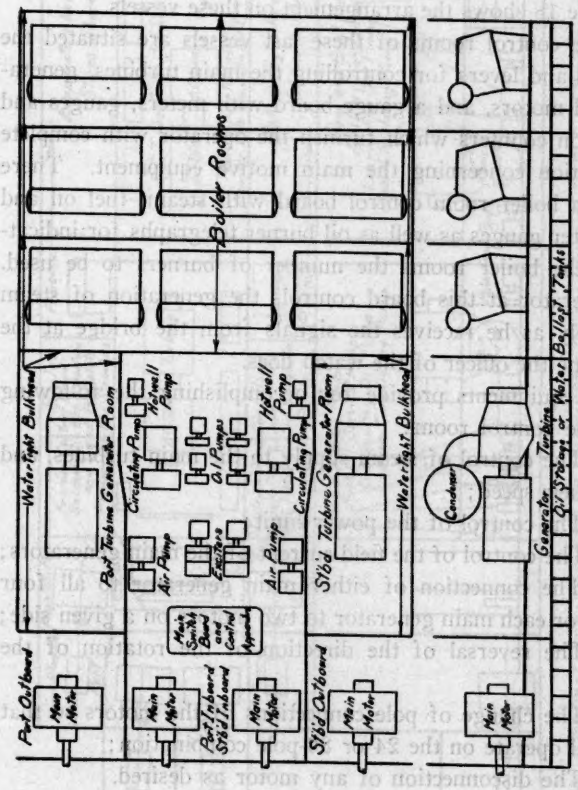


FIG. 17.—PLAN AND ELEVATION OF ENGINE AND BOILER ROOMS ARRANGEMENT U. S. S. "NEW MEXICO."

designed from the beginning as electric drive vessels. The arrangement of their machinery takes advantage of the possibility of dividing the equipment into a number of small units, and we find them much better protected from gun and torpedo-fire than the *New Mexico*.

Figure 18 shows the arrangement on these vessels.

In the control rooms of these last vessels are situated the switches and levers for controlling the main turbines, generators and motors, and a gauge board with meters, gauges and revolution counters which furnish the operator with complete information concerning the main motive equipment. There is also a boiler room control board with steam, fuel oil and feed water gauges as well as oil burner telegraphs for indicating in the boiler rooms the number of burners to be used. The operator at this board controls the generation of steam accurately as he receives the signals from the bridge at the same time the officer of the watch does.

Both equipments provide for accomplishing the following from the control room:

(*a*) The control of steam supply to the main turbines, and hence, their speed;

(*b*) The control of the power limit;

(*c*) The control of the field current of the main generators;

(*d*) The connection of either main generator to all four motors, or each main generator to two motors on a given side;

(*e*) The reversal of the direction of the rotation of the motors;

(*f*) The change of pole connections of the motors so that they will operate on the 24 or 36-pole combination;

(*g*) The disconnection of any motor as desired.

In addition to the above, the following arrangements are made which are applicable to one type or the other:—

On General Electric ships—

Means of changing the alternator connections so that high voltage (4,200) or low voltage (3,000) will be generated;

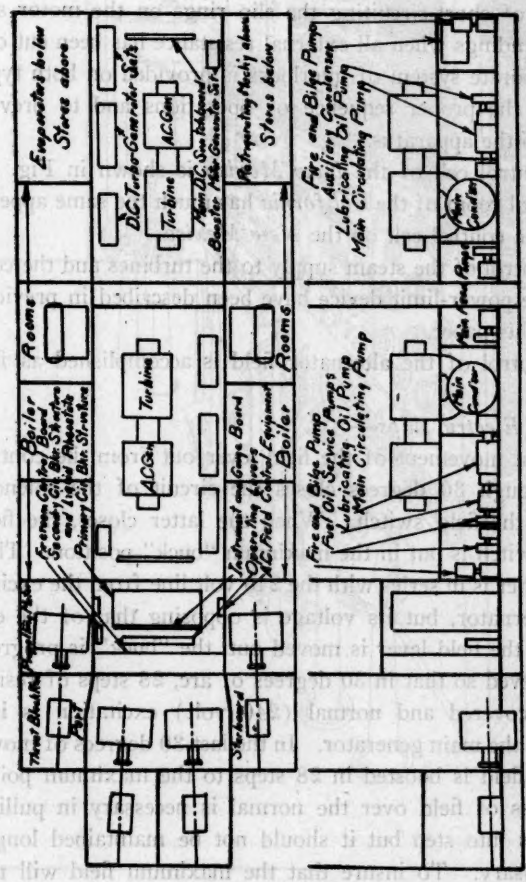


FIG. 18.—GENERAL ARRANGEMENT OF MACHINERY COMPARTMENTS U. S. S. "TENNESSEE" AND "CALIFORNIA."

On Westinghouse ships—

Means of cutting out the external resistance in the motor rotor 24 pole winding as the motors pull into step;

Means of short-circuiting the slip rings on the motor secondary windings when all external resistance has been cut out.

An elaborate system of interlocks is provided on both types to insure the proper sequence of operations and to prevent damage to the apparatus.

The control cell of the *New Mexico* is shown in Fig. 19. The control room of the *California* has much the same appearance as the control cell on the *New Mexico.*

The control of the steam supply to the turbines and the control of the power-limit device have been described in previous pages of this paper.

The control of the alternator field is accomplished as follows:

General Electric ships—

The first movement of the field lever out from the control board through 30 degrees closes the circuit of the solenoid operating the field switch. When the latter closes, the field rheostat switch is put in the maximum "buck" position. That is the booster is in series with the 240 volt line from the exciter to the alternator, but its voltage is opposing that of the exciter. As the field lever is moved out, the "buck" is progressively relieved so that in 30 degrees of arc, 28 steps of resistance are covered and normal (240 volt) excitation is impressed on the main generator. In the last 30 degrees of movement, the field is boosted in 28 steps to the maximum point. This excess of field over the normal is necessary in pulling the motors into step but it should not be maintained longer than necessary. To insure that the maximum field will not be left on, a spring on the field lever returns it to the point of normal excitation when the handle is released. There are no notches on the field lever quadrant between normal and maximum boost excitation postions, so the field cannot be locked in the over-excitation position.

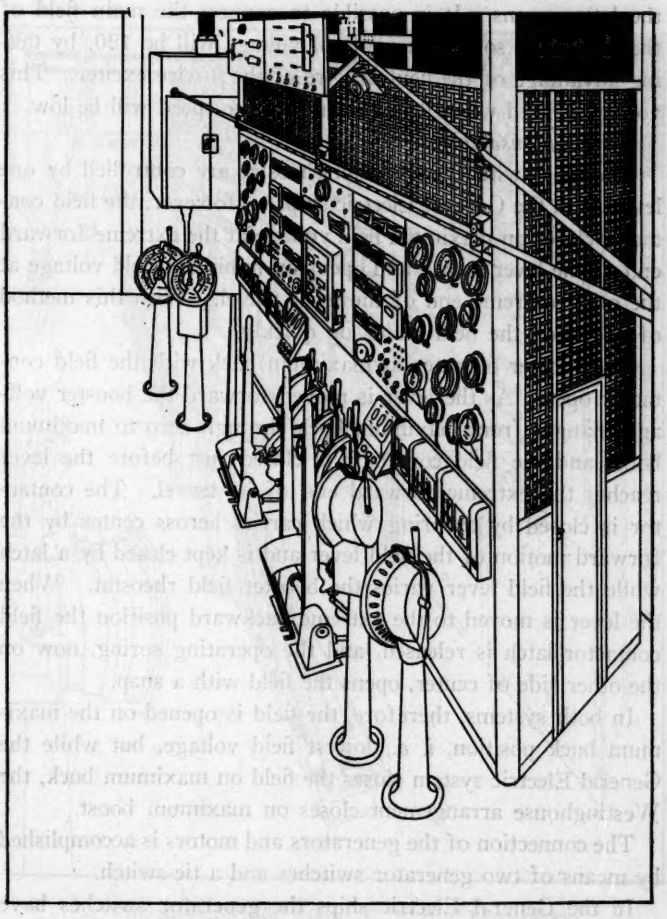


FIG. 19.—CONTROL CELL. U. S. S. "NEW MEXICO."

When the field lever is in the "off" postion, the current to the solenoids operating the field switch is interrupted and the latter opens. It is possible to connect the main field of the alternator so that its normal voltage will be 120, by taking advantage of the neutral wire of the 3-wire exciter. This voltage is used when it is known that the speed will be low.

Westinghouse ships—

The field-switch and booster-rheostat are controlled by one lever as in the General Electric ships. However, the field contactor closes on maximum field voltage at the extreme forward end of the lever travel, and opens on minimum field voltage at the other extreme end of the lever travel. With this method of operation the field builds up quickly.

The booster operates at maximum buck with the field contactor open. As the lever is moved forward the booster voltage changes from maximum buck through zero to maximum boost and the field contactor is closed just before the lever reaches the extreme forward end of its travel. The contactor is closed by a spring which carries across center by the forward motion of the field lever and is kept closed by a latch while the field lever varies the booster field rheostat. When the lever is moved to the extreme backward position the field contactor latch is released, and the operating spring, now on the other side of center, opens the field with a snap.

In both systems, therefore, the field is opened on the maximum buck position, *i. e.*, lowest field voltage, but while the General Electric system closes the field on maximum buck, the Westinghouse arrangement closes on maximum boost.

The connection of the generators and motors is accomplished by means of two generator switches and a tie-switch.

In the General Electric ships the generator switches have three positions—off, high voltage, and low voltage. In the Westinghouse ships the generator switches have but two positions—on, and off. The tie-switch, when on, ties the port and starboard generator buses together when one generator drives

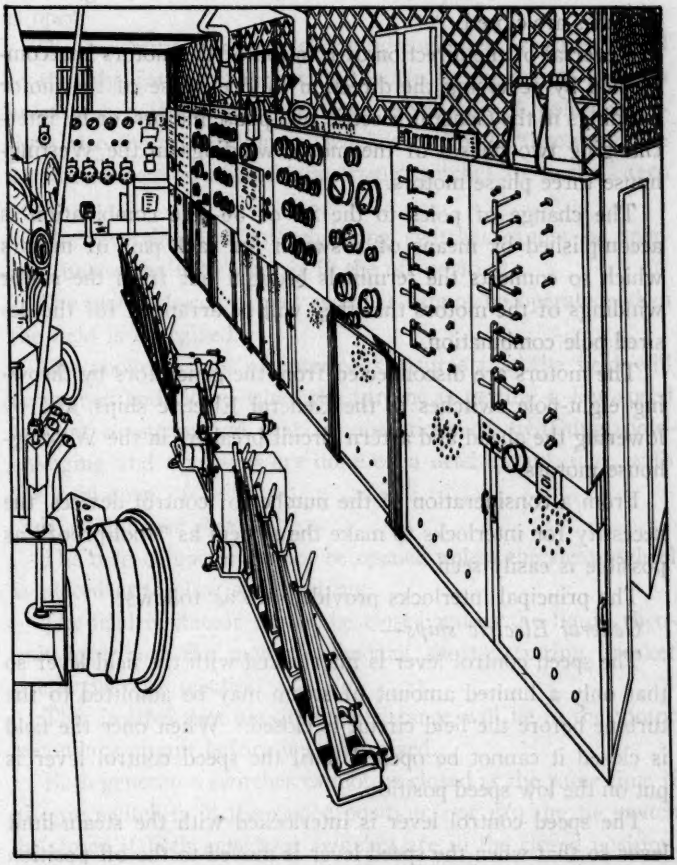


FIG. 20.—CONTROL ROOM. U. S. S. "TENNESSEE."

all four motors. When off, the generator buses are separate and both generators must be running to drive all four motors.

The tie-switch cannot be closed when both generator switches are closed.

Reversal of the direction of rotation of the motors is accomplished by reversing the direction of one phase of the motor windings in the General Electric two-phase motors, or by interchanging two phases of the motor windings in the Westinghouse three phase motors.

The change of poles to the 24 or 36-pole combination is accomplished by means of a switch for each pair of motors which so connects the terminals brought out from the stator windings of the motors that they will be arranged for the desired pole combination.

The motors are disconnected from the generators by throwing eight-pole switches in the General Electric ships, and by lowering the ahead and astern circuit breakers in the Westinghouse motors.

From a consideration of the number of control devices, the necessity for interlocks to make the system as "fool-proof" as possible is easily seen.

The principal interlocks provided are as follows:

General Electric ships—

The speed control lever is interlocked with the field lever so that only a limited amount of steam may be admitted to the turbine before the field circuit is closed. When once the field is closed it cannot be opened until the speed control lever is put on the low speed position.

The speed control lever is interlocked with the steam-limit lever so that when the speed lever is moved to the off position the steam limit lever is tripped to the full value position.

The generator disconnecting switches cannot be opened or closed when energized.

Neither disconnecting switch can be closed when the other disconnecting switch and the bus tie switch are both closed.

The disconnecting switches cannot be closed in the high voltage connection when the bus tie switch is closed, nor can they be closed in the low voltage connection when the bus tie switch is open.

The reversing and pole-changing switches are interlocked so that the former may not be thrown in the astern position unless the pole changing switches are in the 36 pole connection; also the pole changing switches cannot be thrown in the 24 pole connection when the reversing switches are in the astern position.

The reversing and pole changing switches cannot be operated unless the field lever is in the off position.

The motor disconnecting switches cannot be operated when the field is energized.

The above system of interlocks insures that the field will be opened and closed when the turbine is getting a low-speed amount of steam and that all operations of switching, pole-changing and reversing are done on a dead circuit, *i. e.*, with the field open.

Westinghouse ships—

The field contactor cannot be opened unless the steam wheel is placed at the low speed setting.

The field contactor cannot be closed unless the liquid rheostat lever and the motor secondary short-circuiting breaker are in the open position.

This insures that maximum resistance will be in the motor secondary circuit before it is energized.

Both generator switches cannot be closed at the same time if the tie switch is in the closed position, nor can the tie switch be closed if both generator switches are in the closed position. Paralleling the main generators is prevented by the above interlocks.

When the field contactor has been closed all movement of levers is prevented except that of the liquid rheostat lever and the motor secondary lever.

The motor secondary lever cannot be closed until after the liquid rheostat lever has been closed. The motor secondary should not be closed until the liquid in the rheostat reaches its maximum point as observed in the gauge glass on the rheostat tank.

There is no restriction upon the opening of the rheostat lever and motor secondary lever.

When the tie switch is closed, the interlocks of both port and starboard motors are connected so that they are operated by the field lever of the generator in use.

The following pairs of levers are prevented from being in the closed position at the same time:—

Ahead and Back

24 pole and 36 pole

36 pole and Back

The ahead lever cannot be closed until after the 24 pole switch has been closed.

The above interlocks accomplish the same results as those on the General Electric ships—the field cannot be opened unless the steam supply to the turbine is that for low speed and all switching is done with the field de-energized.

AUXILIARIES.

In making up the designs for electrically propelled ships motor driven auxiliaries for the main drive were specified practically throughout. There is little fear today of the failure of electric motors, and their economy over steam driven auxiliaries warrants the slightly greater initial cost. As the generation of power in the alternator depends on the exciter, the auxiliaries may as well be run from the exciter also, for without excitation the ships cannot move.

So in all of our electric drive battleships the following are electric driven:—

Main and auxiliary circulating pumps;

Main and auxiliary hot well pumps;

Forced lubrication pumps;
Oil cooler circulating water pumps;
Main motor ventilating blowers;

The *New Mexico* has in addition, motor driven main air pumps. The latter on the *Tennessee* and *California* classes are steam driven.

On the *Tennessee* class two of the four forced lubrication pumps are turbine driven, arranged to cut in automatically when the oil pressure on the main bearings drops to about 8 pounds.

The *California* class has in addition to her motor driven forced lubrication pumps, gear oil pumps driven by the main turbine shaft.

The above arrangements are provided to protect the bearings of the main turbo-alternators in case of total failure of the electric plant.

In all the electric drive ships the boiler auxiliaries, *i. e.*, fuel oil pumps, main feed pumps, and forced draft blowers, are turbine driven. This permits steam pressure to be raised and kept in the boilers without regard to the electric plant.

The foregoing discussion shows the importance of the exciter sets in the successful operation of the main drive.

On all electric drive vessels one 300 K.W. geared non-condensing three-wire 240/120 volt D.C. turbo-generator is provided for each main turbine. These machines exhaust into the main turbines and so need no separate condensing equipment. They are intended to carry the excitation and main drive auxiliaries alone, and while they can be paralleled with the light and power machines, should not be so operated except in cases of emergency.

The light and power turbo-generators are subject to widely varying loads especially when the turrets are in use, loads that will vary almost instantly from one-third to one-and-a-third of the rated capacity of the machines. The excitation turbo-

generator should not be paralleled to share in such variations of load but should be left alone with the almost steady propulsion load.

In addition to the exciters, the *New Mexico* has four 300 K.W. condensing geared, three wire 240/120 volts turbo-generators installed, two in each dynamo room and intended for the ships light and power. They can be connected to the exciter switchboard through the after distribution room board and so furnish excitation and power for the main drive auxiliaries. They cannot be paralleled with the exciter turbo-generators.

The *Tennessee* and *California* classes also have four 300 K.W. condensing geared three-wire 240/120 volt turbo generators installed, two in each engine room, and arranged with switchboards in all particulars similar to those for the exciter machines. The 300 K.W. generators in each engine room can be paralleled with each other and also with the exciter and any of them can carry the excitation and auxiliary load.

When these machines are paralleled they feed the distribution rooms through their circuit-breakers as do the light and power machines on any of our ships. On each generator board, however, is a main auxiliary bus and a reserve auxiliary bus from either of which power for excitation and main drive auxiliaries may be derived. The main auxiliary bus is connected inside the circuit breakers so that if the latter should open for any reason, the main drive power will not be affected. To this bus are thrown the switches for the booster motor generator and the other main drive auxiliaries.

The reserve auxiliary bus may be energized in two ways—either from the main auxiliary bus in the other engine room, or from the light and power bus in the same engine room. The switches for the main drive auxiliaries are double throw and may be thrown from the main auxiliary bus to the reserve auxiliary bus at the will of the switchboard operator.

On the *New Mexico,* and on the *California* class, an arrangement is provided to energize the field of the main alternator with 120 volt current as well as with 240 volt current. This arrangement does not obtain on the *Tennessee* and *Colorado.*

In all ships the booster generator may be disconnected and the alternator field fed direct from the exciter bus without interrupting the field current. In case this is done, it is necessary to take the other auxiliaries off the exciter so that the main field may be varied by changing the exciter rheostat.

On the *New Mexico* the main circulating pump may be run on 120 volts as well as 240 volts.

On the *California* class, the lubricating oil pumps as well as the main circulator may be operated on either 120 or 240 volts.

The above outline of the excitation and auxiliary connections embodies the essentials of the system for the main drive.

The engine room direct current boards include, of course, a number of connections not here touched upon, but most of them are the same as those found on the generator boards in any dynamo room.

Fig. 21 shows the essentials of the engine room circuits for the main drive.

OPERATION.

The operation of an electric drive ship seems very complicated to one watching the process of getting underway for the first time. When analyzed, however, the manipulation of switches and levers is seen to be logical and simple.

The first set-up depends on whether or not both generators are to be used. This determines the set-up of generator disconnecting switches and the motor tie (bus-tie) switch. On the General Electric ships it also determines whether the high or low voltage connection on the generators will be used.

On the General Electric ships, the 36 pole connection is usually employed in starting the motors and always is in back-

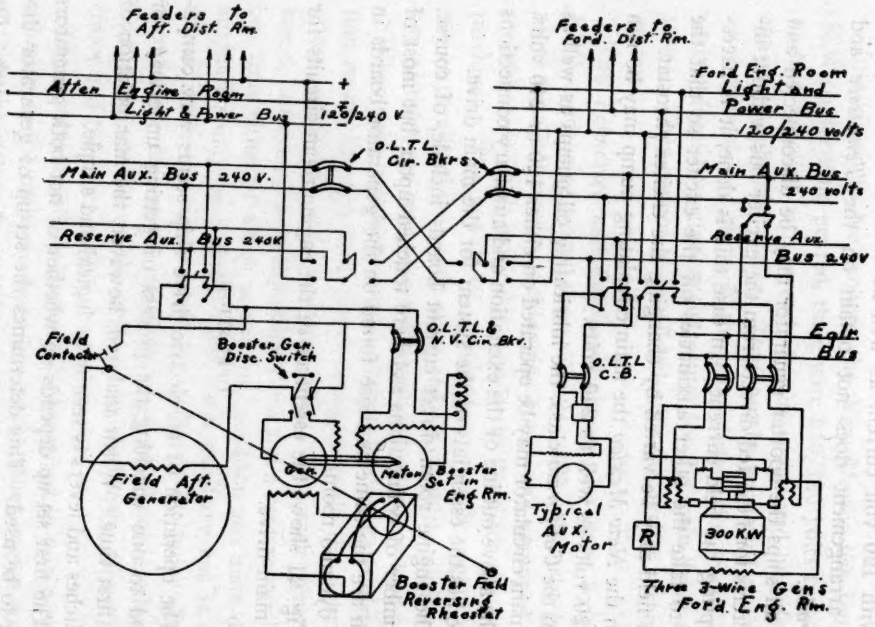


FIG. 21.—ENGINE ROOM CIRCUITS FOR MAIN PROPULSION.

ing, while on the Westinghouse ships the 24 pole connection must be used until the motors are in step, either in going ahead or in backing.

With the proper generator, bus tie and pole-changing switches in, the main turbine at its idling speed, and the field open, the operator is ready for a bell from the bridge.

When a bell is rung up, the "Ahead" or "Astern" switches are thrown, the steam lever opened, and the field closed.

On the *New Mexico*, the field is overexcited until the motors are in step, when the field is reduced to that necessary to keep the motors from falling out of step.

On the *California*, the field lever is moved to the over-excitation point to insure the motors pulling into step. A temporary stop is then lifted and the field lever moved to the end of its travel. This closes the contactors and short-circuits the collector rings on the phase winding. The field lever is then brought back to the normal excitation point.

On the Westinghouse ships the field lever is thrown to the extreme end of its travel. This closes the field contactors and puts maximum excitation on the field. Maximum resistance is in series with the motor secondary winding at this time. The rheostat valve lever is closed allowing the external resistance to be cut out gradually. When it is all out, the motor secondary short-circuiting switch is closed and the rheostat valve lever opened. At the same time the excitation is reduced to the normal amount for the running speed.

On both ships the steam lever is adjusted to give the turbine speed necessary for the revolutions called for by the bridge.

If, after starting, it is desired to change from one pole combination to the other, the procedure is as follows:

On General Electric ships—

When the motors have pulled into step after starting on the 36 pole connection, bring speed lever to idling position and field lever to open position. Throw pole changing lever to 24 pole position, and proceed as in starting on the 36 pole connection.

All that is necessary to do then is to bring the speed lever to slow speed, open the field and move the pole changing switch as rapidly as possible. Then close field and open steam.

On Westinghouse ships—

All starting is done on the 24 pole connection and no attempt to shift to 36 poles should be made until the ship is up to the speed corresponding to the propeller revolutions. Then set steam wheel at idling position, open field, pull out motor secondary and 24 pole switches, put in 36 pole switches close field and open steam wheel to proper position for revolutions desired. This operation can be performed in about three seconds with a loss of but thirty revolutions.

The usual set up of switches for the various speeds is as follows:

Speed in Knots	No. of Generators	No. of Motors	Pole Connection on Motors	Bus Tie Switch	Generator Connections (General Elec.)
0-15	1	4	36	closed	low voltage
15-17	1	4	24	closed	low voltage
17-Full	2	4	24	open	high voltage
Backing	1	4	36 G.E. 24 W.	closed	low voltage
	2	4	36 G.E. 24 W.	open	high voltage

The generator connections on the Westinghouse machines are never changed.

OPERATING PRACTICE.

The following instructions for getting underway, coming to anchor, standing by, etc., are taken from operating instructions on electric drive vessels. The practice on the *Tennessee* is taken as typical of the Westinghouse installations, while that of the *New Mexico* will represent the practice on General Electric ships. However, the *California* and *Maryland* have their boilers, auxiliaries and steam lines laid out in a similar manner to those on the *Tennessee,* so that much of the practice in use on the *Tennessee* is also applicable to the *California* and *Maryland.*

I. PROCEDURE IN GETTING UNDERWAY—U. S. S. "TENNESSEE."

(*a*) Preliminary Operations.

If sufficient notice is given about time of getting underway, the following should be done the day before:

Ascertain what the standard speed will be for getting underway and prepare sufficient boilers to give necessary power. Get them ready for lighting off.

Inspect bilge strainers, fuel oil and lubricating oil strainers.

Clean oil wells of bearings of fuel and main feed pumps and renew oil.

See sufficient lubricating oil run down from the storage to the drain tanks.

Examine main turbines and auxiliaries, examine main throttle valve to see that it can be opened, operate main turbine nozzle valves.

See that Engineer Department in general is ready to get underway.

Warm up main set.

Three hours is sufficient time to get underway from the time of lighting off boilers.

See that fuel oil, main feed, and blower steam and exhaust valves to the boilers are open.

Light off boilers, bring up slowly, carrying out regular routine for lighting safeties, etc., and cut in very slowly when ready. The short steam lines in this vessel necessitate special care in this matter.

When two boilers are on the line all preparations can be carried out up to rocking the main motors.

In the engine room have two 300 K.W. condensing turbo-generator sets cut in to give all necessary power.

Split feed and parallel fuel oil system between forward and after engine rooms. Starboard boilers feed from forward system, port boilers from aft; in case of emergency any boiler can shift to the other system. Two fuel oil pumps, one forward and one aft, on the fuel oil system, cross-connection open.

Crack valves on condenser heads to be sure that the condenser is not air bound.

Start electric driven forced lubricating oil pump and when pressure is on the system test out steam driven pump by tripping out the electric pump. The steam pump should start automatically when the lubricating oil pressure falls to about 8 pounds, and should take up the suction immediately.

Open the drains on the main turbine and steam chest and crack the by-pass to the main steam chest.

Warm the main air pump, start and run slowly with the vacuum breaker open.

Clear condenser of water.

Open the by-pass gradually and bring the pressure in the main turbine steam chest up slowly.

Remove the block holding up the governor inlet valve. (This block is put in when securing to keep the valve from freezing shut.) Try out the governor control and power limit device, leaving the power limit indicator at about "50" for starting up. Put block back after trying out the governor.

Check the thrust setting.

When the pressure on the steam chest has reached the main steam line pressure, crack the main stop and gradually open the valve to four turns.

Close the by-pass.

About five minutes before starting the main turbines, turn on the gland steam keeping enough pressure to blow steam out along the shaft, close the vacuum breakers and bring the vacuum up to 12 or 14 inches. Open one nozzle, No. 3 preferably as this will usually be the operating nozzle, and should always be used in getting underway.

Open throttle quickly to about 50 pounds on first stage until turbine starts, then close. Keep the turbine turning and listen for gland rubs. Give the turbine steam every two or three minutes and bring up to idling speed.

Close turbine drains.

TESTS JUST PRIOR TO GETTING UNDERWAY.

Prior to taking over control in the control room the officer of the watch will inspect the relay pumps and tanks and make sure the floats are not sticking.

Jack motors one and a quarter turns and disconnect jacking gear.

Turn on oil to motor and thrust bearings. Open circulating water supply to shaft alleys and thrust bearings.

Test out rheostat pumps and make sure that the rheostat water is flowing properly.

When the engine room is ready, take over control of the main turbine, close speed wheel (this has been opened about two notches), have block taken out from under governor valve and bring power limit down to about "ten." (This has been at about "fifty.")

Have the booster set started in the engine room.

Test out turbine speed control from the control room by opening speed wheel gradually, watching to see that the first stage pressure mounts up at the same time and turbine speed increases. Get report of this from the engine room and whether or not main throttle valve, gland water and power limit operate properly. The main turbine vacuum should be from 20 to 24 inches when ready to move the main motors.

Warm up non-condensing 300 K.W. set and at 20 minutes before getting underway, cut in and take over excitation and main drive auxiliaries.

Shift auxiliary exhaust to turbine. Regulate back pressure to 6 or 8 pounds.

Throw in 24-pole switches, motor tie switch (if one generator) and generator switch for generator in use.

Start rheostat pumps.

When permission is granted by the Officer of the Deck, rock the main motors by putting in direction levers (one ahead and one astern on each side) then throw in field lever all the way,

and then the rheostat levers. As soon as the motors start, pull the field, rheostat, and direction levers. Reverse direction levers and repeat.

Report to the officer of the deck ready to get underway.

Keep turbine idling at 600 to 700 R.P.M.

(*b*) Maneuvering.

The usual condition of getting underway will be on one main generator and four boilers. Number three nozzle will give sufficient steam for fifteen knots and is used in maneuvering.

Keep steam pressure on boilers between 240 and 250 pounds while maneuvering so that sudden stop will not cause the safety valves to lift.

With number three nozzle open, the position indicator should be set at ten (10) for starting from stop. With this setting the throttle may be opened quickly, without danger of carrying over water from boilers.

When a bell is received (with position indicator at 10) open speed wheel to third or fourth notch and adjust speed to that required with power limit, running it up or down depending on what speed is required.

The field lever should be thrown all the way in when a bell to go ahead or astern is received and the field then gradually reduced to the proper strength for the speed required. As soon as the field is in, throw over rheostat valve lever, keep it in until the water reaches square marks on tank gauge glasses, then put in the motor secondary short-circuiting switch lever and pull out the rheostat valve lever. Carry plenty of field when getting underway so that the motors will not drop out of step.

When "Standard Speed" is received get up to it as quickly as possible as follows:—Open speed wheel to fourth or fifth notch and come up quickly on power limit until speed is reached. At the same time increase the field excitation to keep a stable condition. As soon as speed is reached, ease down on

the speed wheel so that the turbine is running on the governor, making the proper speed. Then lower power limit so that it is just above the governor (red light flickering on and off). Field may now be cut down.

For changes in speed formation proceed as follows: Open or close speed wheel until it is on proper setting for speed desired and then move power limit up or down until it is just ahead of the governor.

Always watch the excitation to see that stability is maintained.

The proper position of the speed wheel for given number of revolutions should be known or obtained very quickly and there should be no trouble in coming to a given speed.

The function of the power limit device is to limit the amount of steam taken by the turbine and its primary function is to keep the motors from dropping out of step when the rudder is put over.

2. OPERATING INSTRUCTIONS.

(*a*) Machinery in Operation for Various Speeds.

(1) Main Turbine and Main Drive Auxiliaries.

As soon as the ship has straightened out and "Standard Speed" is being made, shift from main air pump to the condensate pump and air ejector and shut down main air pump. Set the relief valve to the main condenser set at 10 pounds and the back pressure valve on the auxiliary condenser at 13 pounds. Keep two 300 K.W. condensing machines running at all times on ship's light and power. One ventilating blower on each main motor is sufficient at cruising speed. The speed of the blower should be varied for various speeds.

When making over twelve (12) knots the fuel oil booster pump should be used to feed oil from the belt line to the fuel oil service pump. At least two fuel oil pumps will be kept running at all times.

Great care must be taken to keep up superheat temperature and to keep down vacuum temperature.

Water should be kept below four inches in the boiler gauge glasses.

The speed of the main circulating pumps and the number of air ejectors in use should be regulated to obtain the most economical condition. The extra steam consumed by reason of ejector steam and extra current for the main circulating pump should be checked against the steam saved by higher vacuum.

(2) Main Generators and Main Motors.

Always, in starting main motors from a stop and when maneuvering, the 24-pole combination is used. There is no starting resistance in the 36-pole winding and any attempt to start up on this combination would result in serious damage to the windings. Therefore when it is desired to shift to 36-pole combination the ship must have sufficient way on to keep the propellers turning over while the shift is being made. The speed ratio of the 36-pole connection is approximately 18 to 1 and so to prevent overspeeding the generator, only speeds up to 15 knots can be made on the 36-pole combination. For speed beyond that the 24-pole combination is used.

The field excitation must be carefully watched at all times to prevent the motors from dropping out of step. This is particularly true in maneuvering where sudden increases in load are frequent and the field should be over-excited, with power factor of about 60, to guarantee stability. When steaming steadily at constant speed the field should be reduced as low as possible within limits of stability, carrying a power factor of about 75, thus keeping the field temperature down.

If when steaming, a sudden overload causes the motor to drop out of step, the following procedure should be followed:—

24-Pole Connection—Pull out motor secondary lever, increase field, start up rheostat pumps and throw rheostat levers. When water has reached upper level throw in motor secondary and pull rheostat levers. Build up to speed again.

36-Pole Combination—Here it is necessary to pull field lever and 36-pole levers and start up again on 24-pole combination.

(3) Auxiliaries.

Boilers in Use—Sufficient number to give maximum required power with size tips to give maximum economy.

300-K.W. Turbo Generator—Under normal steaming conditions using one main generating unit two 300 K.W. turbo generators will be kept on the ship's light and power and one generator on the main drive auxiliaries and excitation. The non-condensing machine carries a back pressure of about 10 pounds, part of which goes to the feed heater, part to the evaporators and the remainder to the second stage of the main turbine.

The usual operation of the 300 K.W. machines is for the non-condensing machine to be used for excitation and main drive auxiliaries and for no other purpose. Other machines operate on ship's light and power. Since the main propulsion of the ship depends upon the proper operation of the non-condensing machine great care must be taken to insure proper operation of this machine. Considerable trouble has been experienced with the primary valves of these machines sticking, causing the machine to speed up or slow down with consequent danger of tripping out. The primary and secondary valves should be worked by hand frequently during each watch to prevent sticking and the supplementary steam valve should never be opened unless carrying an unusually high steady load around 1,500 ampères.

Ordinarily one 300 K.W. machine can carry a constant load of 1,400 ampères and peak leads of 1,600 ampères without danger of tripping out but if the load exceeds the above amounts another machine should be cut in. It is more economical, naturally, to run one machine at high load than to divide the load between two machines but the danger of tripping out, with the consequent loss of power, overbalances the

small amount of oil saved by one machine at high load and for safety reasons two machines are kept on ship's light and power at all times.

Fuel Oil and Main Feed Pumps—These pumps are used as described before, at least two of each always to be used.

Lubricating Oil Pumps—If operating on the after engine room only the after electric driven pump is used, with the steam pump as a standby, this pump supplying the motors and turbines and the hydraulic speed control system. However, if operating on forward engine room, both the forward and after pumps must be used, as forward pump does not supply the motors and shaft alleys. Eighty pounds pressure is carried on the hydraulic system and 12 to 15 pounds on the bearings.

U. S. S. "NEW MEXICO."

Have two ship's dynamos running and cut in by the time fixed to start warming up (not less than 1½ hours before time for getting underway). Have main feeder from after distribution board to exciter switchboard cut in. Open main injection and overboard discharge. Crack by-pass on main steam stop. Start main circulating pump on low voltage (slow speed). Bring pressure up on main steam line very slowly, allowing at least one hour for warming up before main stop is opened. Pump out main turbines with turbine drain pumps. Start condensate pump. Turn on steam to gland seals and notify generator rooms to seal glands. Start dry air pumps. Start lubricating oil pump supplying oil to main turbine governors and bearings, and bearings of main motors—70 pounds to governor, 20 pounds to generator bearings, and 30 pounds to motor bearings. After about 25 inches of vacuum has been obtained, the auxiliary exhaust may be shifted from the dynamo to the main condenser, and the 300 K.W. exciter sets started. After they are up to speed with proper voltage built up (240 V.), the load of the engine room auxiliaries may be shifted from the after distribution board busses to the exciter

busses. When the by-passes on the main steam stops have been open a sufficient length of time (1 hour), to warm up the steam lines and main turbines, open the main steam stops. Open the water service to generator bearings from main circulating pumps. After inspecting main turbines and generators, if it is found they are receiving the proper amount of lubricating oil and bearing cooling water, they may be started. The main throttles should be opened only a sufficient amount to start the machines and bring them up to speed (700 R.P.M.), to make sure the governor is functioning properly. At this time, the throttles should be tripped by the emergency trips from the main control board. If they function properly, throttles should be reset and again opened a fraction of the way, then the governor control from the main control board should be tried out, and if turbine controls properly, main throttles may be opened wide.

Start motor generator booster sets and try out main field switches with maximum excitation. Inspect main motor and thrust bearings for proper amount of oil. See that jacking gears are secured. Inspect high tension cell and high tension leads. Close bus tie switch, all motor disconnecting switches, and one main generator switch on low voltage. Close both pole changing switches on 36 poles, one reversing switch astern, the other reversing switch ahead. Move main generator field control lever to maximum excitation position. As soon as the motors turn over, shut off the excitation, then throw the reversing switches to the opposite position, proceed as before and turn over the main motors in the opposite direction. Upon completion of this, open the generator switch just used and close the other generator switch on low. Now proceed to turn over main motors as described above. Start main boiler feed pump. Test out and notify fire room to secure auxiliary feed pump. It is advisable to start a main feed pump and shift to main feed lines about ten minutes before trying out main motors.

Auxiliaries (*New Mexico*).

The *New Mexico* keeps two ship's dynamos in either the forward or after dynamo room in use in addition to one exciter in the engine room for each main generator in operation. Normally, the exciter furnishes energy for the engine room auxiliaries and for excitation, but in case of emergency, these auxiliaries can be thrown over to the ship's dynamos, and the main generator field can be excited from the same source. The exciters in the engine room exhaust into the fifth or eighth stage of the main turbines, or into main condensers.

The proper speed, to obtain the desired amount of water through the main condenser, is determined by comparing the temperatures of injection, overboard discharge, and condensate pump discharge. The best condition is obtained with a rise in temperature of about 14 degrees F. between injection and overboard discharge, and about 7 degrees F. difference between overboard discharge and condensate pump discharge. The condensate temperature should also be about 3 or 4 degrees F. below the theoretical steam temperature corresponding to the vacuum in the condenser. If the temperature difference between injection and overboard discharge is less than 14 degrees F., the speed of the circulating pump may be reduced.

The condensate pumps are provided with vents from the top of each main condenser. These vents place the vacuum of the condenser on the suction chamber of the pump. The water can then flow into the impeller blades by virtue of the head on it from the bottom of the condenser. Any appreciable leak in the suction lines or vent lines to this pump will cause the pump to lose suction. Leaks in the vent line may be detected by a frost formed on the pipe, since the air expanded from atmospheric pressure to a low pressure will absorb heat in the process. A loss of suction of the pump would be indicated by a decrease of current drawn by the motor, which would be indicated by the ammeter in this circuit on the exciter

switchboard. A gauge glass on the side of the condenser indicates the amount of condensate in the condenser and this glass should be watched carefully.

3. COMING TO ANCHOR AND SECURING (U. S. S. "TENNESSEE."

a. Coming to Anchor.

On receiving first bell to slow, start main air pump and secure the condensate pump and air ejectors. Open No. 3 nozzle on the main turbine, if it is not already open, so that power will be available for backing. Watch back pressure when maneuvering for the anchorage so that it will not get above 15 pounds and lift the relief valves on the forced draft blowers.

b. Securing.

1. At first bell to reduce speed, start main air pump and cut out air ejectors and condensate pump. Always have No. 3 nozzle open when coming to anchor so that plenty of power for backing may be available.

2. Upon orders to secure proceed as follows:—

(*a*) Shift main auxiliaries to reserve bus.

(*b*) Trip out and secure non-condensing machine. Shift auxiliary exhaust from main turbine to auxiliary condenser.

(*c*) Trip main turbine from control room. Close main throttle and steam stop to main turbine.

(*d*) Trip out booster set.

(*e*) Secure water to turbine glands.

(*f*) Close all nozzles to main turbine when steam chest reads zero.

(*g*) Open vacuum breaker slowly—maintaining 5 inches vacuum for a period of at least 15 minutes after turbine is tripped out.

(*h*) Secure main air pump when condenser is cool and clear of water.

(*i*) Run main circulator at slow speed.

(*j*) As soon as turbine rotor stops secure forced lubrication pump and 15 minutes later secure main circulating pumps.

(*k*) Start auxiliary feed and oil cooler pumps.

3. Control Room—Shaft Alleys.

Upon receiving orders to secure:—

(*a*) Trip main turbine from control.

(*b*) Close off oil supply to motor and shaft bearings.

(*c*) Secure after lubricating oil pump if operating with forward engine room.

(*d*) Pull all motor blower switches.

(*e*) Secure water supply to bearings aft.

(*f*) Secure all boilers except one designated for auxiliary purposes.

(*g*) Pull all lever switches out.

(*h*) Take counter readings of all shafts.

(*i*) Trip out vibrator motors.

(*j*) Put shields on all motors if motors are to be idle for more than twenty-four hours.

(*k*) Set up on stern glands.

U. S. S. "NEW MEXICO."

When word to "Secure Main Engines" is received:

(1) Trip the generator throttle by hand and close the main steam stop.

(2) When turbine has stopped, de-energize main field switch and interlocking system and stop booster. A moderate field may be applied to slow down the generator and bring it to a stop.

(3) Open generator disconnecting, bus-tie, and main motor switches.

(4) Shift all motor driven auxiliaries to after distribution board bus. Run the circulating pump motor at low speed on low voltage.

(5) Secure exciters.

(6) When main stop is reported closed, secure oil pump, gland steam, dry air pump, oil cooler pump, and salt water drain pump. Secure cooling water to bearings.

(7) Shift main feed system and secure all unnecessary boilers in firerooms. The most desirable plan is to retain two boilers on the line until securing is finished. Then keep only one auxiliary boiler.

(8) Open air ducts leading to generator and air shutter exhaust trunk.

(9) When exhaust trunk is sufficiently cooled, secure condensate and main circulating pumps, and main injection and overboard valves.

(10) Close up generator air supply and discharge. Turn on drying heaters under generators.

(11) Take up on stern glands and secure shaft alleys.

4. PROCEDURE WHILE STANDING BY.

While standing by, the steam pressure should be dropped 25 or 30 pounds below the normal pressure used underway, and the feed water pressure reduced a corresponding amount. Slow down the main circulator so that the difference in the injection and overboard discharge temperatures will be about 14 degrees F.

5. PROCEDURE AT ANCHOR.

While at anchor, the following routine tests and inspections shall be carried out:—

Daily.

1. Jack all four shafts 1/3 turn every day.
2. Run one main dry air pump 15 minutes every day.
3. Run one oil pump on whole forced lubrication system for 30 minutes every day while shafts and generators are being jacked.

4. When ship remains at anchor for more than three days, jack over each main turbo-generator 1/3 turn every day.

5. Pump both main turbines with drain pump before and after running dry air pump.

6. Operate governor mechanism on turbines by oil pressure.

7. Inspect the motors and generators for moisture including ventilating ducts, air pockets under the generator and fan housings over main motors.

8. Move all motor and generator disconnecting switches.

9. Move all pole changing and reversing oil switches.

10. Operate field control switches.

11. Take megger test of insulation resistance of generator and motor stators and generator field.

12. Keep sufficient heat on main motors to maintain a temperature of 6 degrees above room temperature.

13. Jack over all idle machinery.

Weekly.

1. Make insulation tests of all windings, including bus structure and cables.

2. Make insulation tests by a bridge megger of all electrical circuits and machines.

3. Operate idle valves and operate all shutters in ventilating ducts of main generators and motors.

4. Move all auxiliaries by power.

5. Operate exciter and test overpseed trip and back pressure valves.

6. Examine and test oil in service tanks for water and sediment.

Monthly.

1. Examine stator coils of main generator and motor for dryness and preservation of insulating material. Inspect for presence of oil or salt, or cracking of varnish.

2. Examine by test, all bus bars, securing bolts and connections.

3. Examine all leads and connections of instruments and instrument transformers on both the alternating and direct current boards.

4. Inspect all circuit breaker parts of all main motor breakers.

5. Draw oil from all auxiliary motor bearings and renew.

Quarterly.

1. Take bridge gauge readings of main generators and air gap measurements of main motors.

2. Clean generators and motors by compressed air; varnish if necessary.

3. Overhaul collector rings, brush riggings, brushes and commutators; check spacings and angle of bruhes.

4. Go over contact adjustments of all main switches with feelers.

5. Go over holding down bolts on generator and motors and stay bolts on motor rotor.

6. Take turbine wheel clearances.

7. Remove main turbine governor casing and inspect mechanism for condition and adjustment.

Semi-Annually.

1. Inspect resistance grids.

2. Inspect flexible jaw coupling between turbines and generators.

3. Overhaul self-closing throttle valves.

4. Open and inspect condensers for dirt and corrosion.

5. Open exhaust trunk. Inspect interior and last stage turbine.

6. Inspect steam strainers in main turbine throttle valves.

7. Inspect water service lines to turbines for deterioration.

8. Inspect and take clearances on tail shaft and stern tube bearings if dry-docking period is available.

Annually.

1. Remove all oil from switches and compensators and conduct dielectric test and renew or filter. Determine flash point and viscosity of oil.

2. Check readings of all AC and DC meters and send to Navy Yard for calibration and repair.

3. Spray all stator and connections with high grade insulating varnish.

4. Break down, clean and re-insulate with air drying or baking varnish all auxiliary motors that have been exposed to excessive moisture or dampness.

NOTE: In compiling this article the following publications were consulted:

JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS.
General Electric Review.
Electric Journal.
Electric Propulsion, Comdr. S. M. Robinson, U. S. N.
Alternating Current Electricity, Timbie and Higbie.
Naval Electricians Textbook, Capt. W. H. G. Bullard, U. S. N.

LOW PRESSURE VACUUM EVAPORATORS.

(MULTIPLE-EFFECT FILM TYPE).

By Commander C. A. Jones, U. S. N.,

AND

Lt. Commander A. M. Charlton, U. S. N., Members.

Until quite recently the evaporators used in the Navy have resembled fire tube boilers using high pressure steam for fuel. The evaporator shell contained a comparatively large body of salt-water in which was immersed the steam coil. This coil, into which the evaporating steam was led, took on quite a variety of shapes. Some coils were made up of horizontal U-tubes with one or several passes, some had a number of vertical spirals in parallel in which the steam entered at the top, condensed and was led away at the bottom and others had horizontal straight tubes with front and back headers.

In all these types the steam was taken direct from the boilers and reduced to various working pressures up as high as 150 pounds. The evaporation of the sea water took place at atmospheric pressure or slightly above and was led to the distillers or auxiliary condensers where it was condensed into water.

Great trouble was experienced in these evaporators with scale forming on the tubes. Starting with clean tubes it was necessary to raise the steam pressure day by day until to get the capacity of the plant the limiting pressure on the coil was reached. At this point the plant had to be shut down, the coils pulled out and the scale removed from the tubes. This scale was very difficult to remove so that the operation of scaling was very tedious and frequently resulted in injury to the tubes.

With this type of plant little use could be made of the auxiliary exhaust as with the low pressure available sufficient water could not be produced to render the ship self-supporting.

Various experiments were made using an air pump on the vapor side so that the sea-water would be evaporated at a lower temperature and thus increase the capacity of the plant when operating on auxiliary exhaust.

In 1912 an evaporator of an entirely new type was installed on the *Dixie*. Another plant of the same type was installed on the *Salem* in 1913. Although these plants were quite satisfactory and produced fresh water of excellent quality no further installations were made on Naval vessels until the *Tennessee* and *California* were commissioned in 1920 and 1921. The success of these plants together with the constant failure of other types of evaporators to operate satisfactorily led to the adoption in 1922 of this type of evaporator as a standard for Naval vessels.

The adoption of this standard has resulted in specifying it for new construction as well as the conversion of existing plants, to operate on the new principles involved. The conversion is projected for all ships of the Navy as funds and time permit.

The new standard type of evaporator is based on the patents of Mr. S. Morris Lillie with modifications designed by the Bureau of Engineering and will be known as the modified Lillie evaporator.

These evaporators will use low pressure steam and will be built in multiple effect (double, triple or quadruple). An air pump will be used on the distilling condenser so that the evaporation in the last effect shell will take place in 26 inches or 27 inches of vacuum. There is nothing new in the above arrangements, all of which have been used before. The method of circulating the salt water feed for evaporation within the shell, however, is entirely different from that at present in use.

Instead of having a drowned tube effect with a large body of water in the shell from the surface of which steam is disengaged, the Lillie evaporator uses "film" evaporation or more properly "rain" evaporation.

The feed entering the system is sent through all of the effects in series finally being discharged overboard with a salinity of 2.5/32ds (2½ times normal salinity of sea water). At each effect a circulating pump takes a suction from the bottom of the shell. Part of this water is discharged to the next effect. The remainder of the water is pumped back into the shell to a point above the tube nest where it is discharged over a perforated distribution plate. From this plate the water descends in a rain of drops over the tube nest. Part of each drop evaporates and the rest of the drop washes the salt remaining from the evaporation off the tubes down to the bottom of the shell.

The water level in the bottom of the shell is kept just above the lower row of tubes.

With the above method of circulation, "flash" evaporation takes place, there being sufficient extra feed to keep scale from forming in any great quantity upon the tubes.

The advantages of the above method of evaporating fresh water are many, of which the following may be mentioned:

(1) Use of all auxiliary exhaust steam available.

(2) By using low pressures (which means low temperatures) the possibility of the deposit of sulphates on the tubes is practically obviated. This deposit is exceedingly hard and forms a scale which is most difficult to remove. It has been found by experiment that sulphates are precipitated and commence to adhere to the tubes at a temperature of about 265 degrees F. Pressures of 10 pounds gauge (and below), therefore, are quite safe as far as sulphate precipitation is concerned. Below 265 degrees F. only carbonate scale is formed and the amount of this decreases rapidly with decrease of temperature and this scale ceases to form at about 195 degrees F.

(3) By using a high vacuum the temperature range between the entering steam and the last effect vapor is large even with pressure below 10 pounds gauge.

This temperature range allows the plant to be arranged in triple and quadruple effect, whereas the high pressure drowned tube types were never designed for more than double effect. By the use of multiple effect evaporation the economy is much better than with single effect. Under similar conditions of steam to first effect, vacuum in last effect shell and heating feed water as much as possible with the plant's own available heat, and in every case rejecting the brine from last effect at about 2½/32, we may expect about the following results:

Single effect— .95 pound of dstilled water per pound of steam.
Double effect—1.60 pounds of distilled water per pound of steam.
Triple effect—2.50 pounds of distilled water per pound of steam.
Quadruple effect—3.30 pounds of distilled water per pound of steam.

With a constant amount of steam available, which is practically the condition on any one vessel at anchor, the capacity of the evaporating plant is much increased by multiple effect operation.

The details of an evaporating plant of the Lillie type will now be taken up and the flow of steam, salt water and vapor traced through the plant.

In the figures below, the various systems of piping for the evaporators are shown. The systems are shown separated so that there will be no difficulty in tracing them out.

Figure 1 shows the steam piping, figure 2 the salt water piping, figure 3 the vapor and fresh water piping, and figure 4 the piping for gland sealing and priming of pumps and air removal from steam coils. Where there are variations from the diagrammatic plans, they are necessitated by the layouts of particular vessels.

Steam for the evaporators is obtained from the auxiliary exhaust line or from the auxiliary steam line. Ordinarily there will be sufficient auxiliary exhaust available to evaporate

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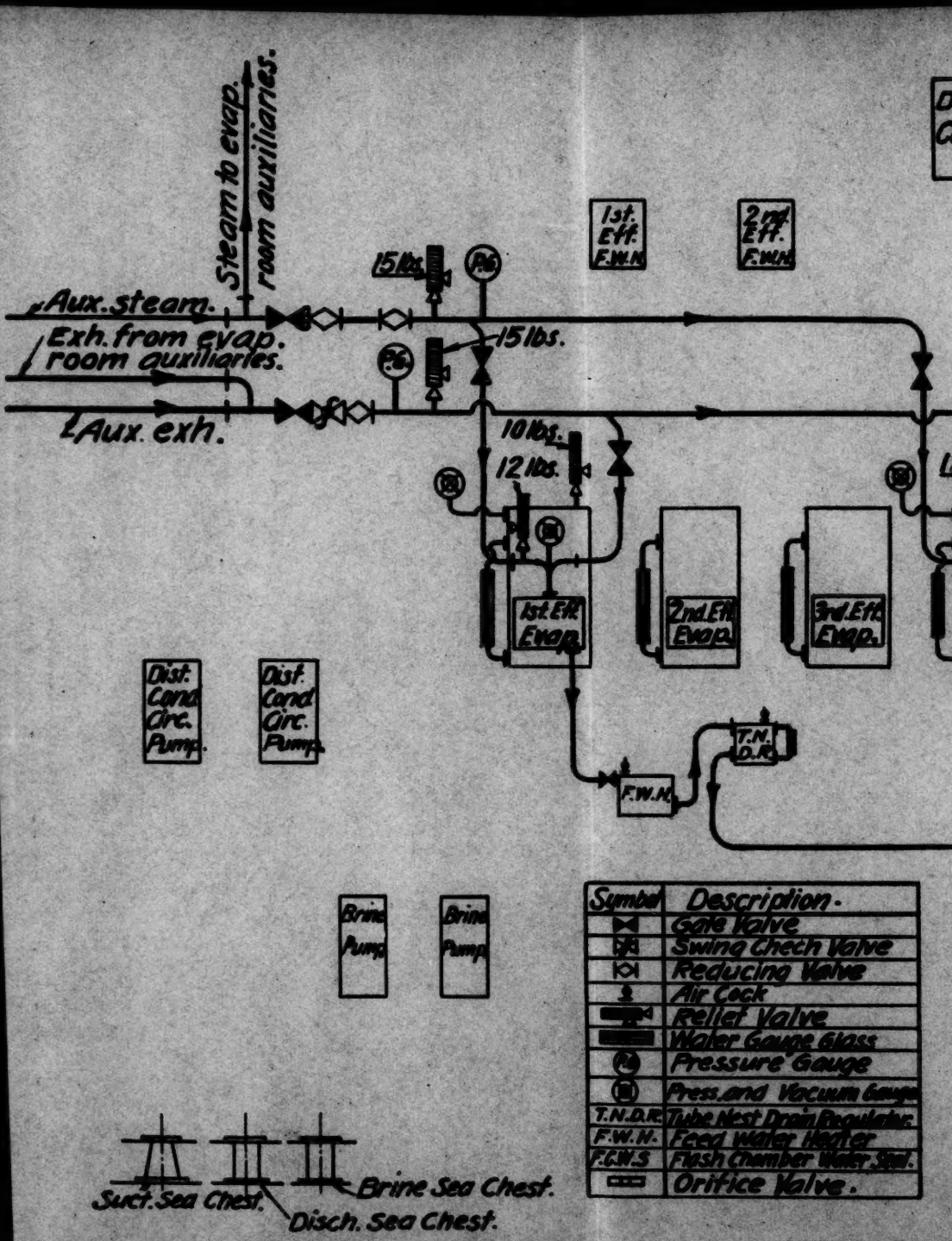
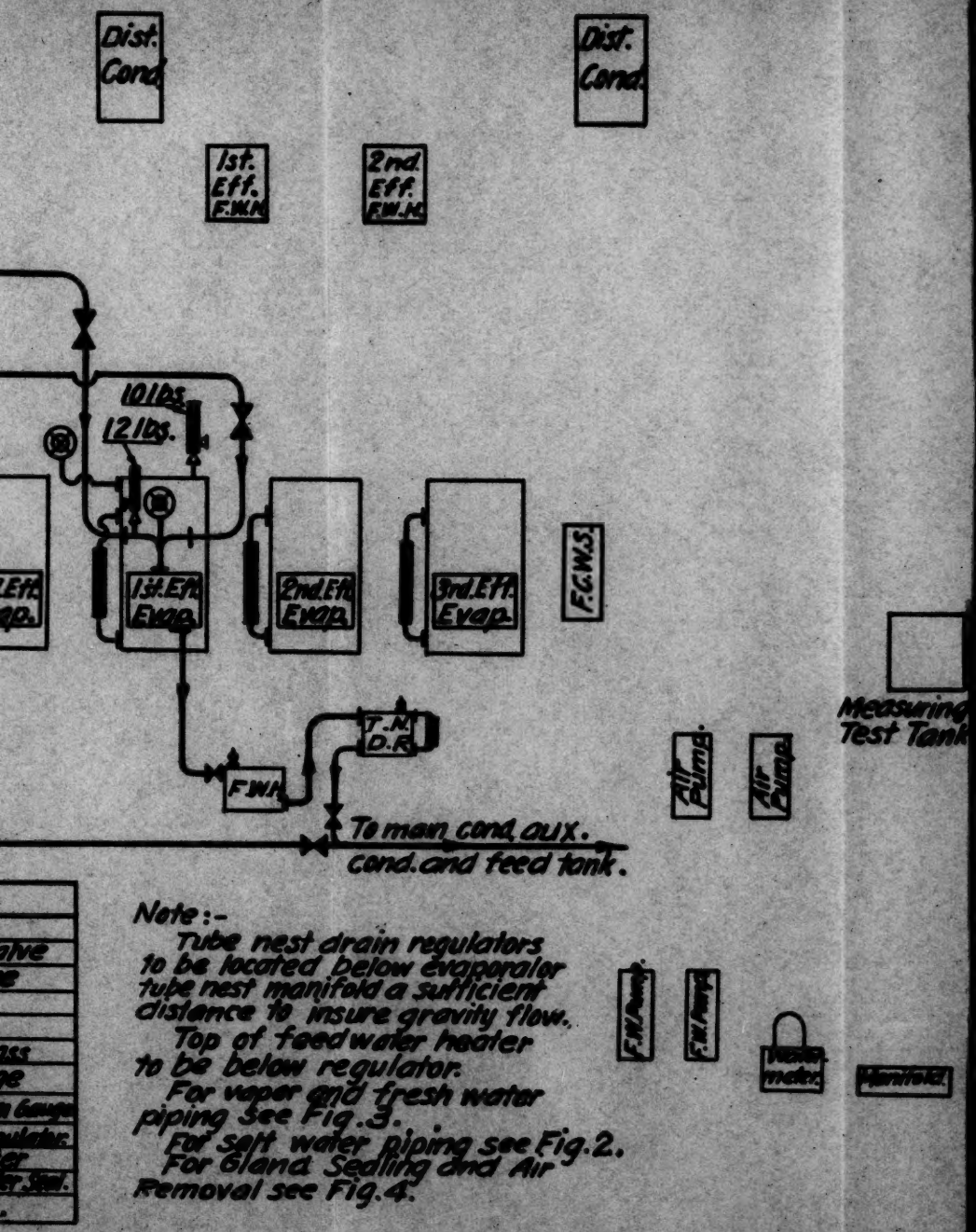


FIG. 1.—STEAM LINES AND FIRST E



Note:-
Tube nest drain regulators to be located below evaporator tube nest manifold a sufficient distance to insure gravity flow.
Top of feed water heater to be below regulator.
For vapor and fresh water piping see Fig. 3.
For salt water piping see Fig. 2.
For Gland Sealing and Air Removal see Fig. 4.

AND FIRST EFFECT TUBE NEST DRAIN

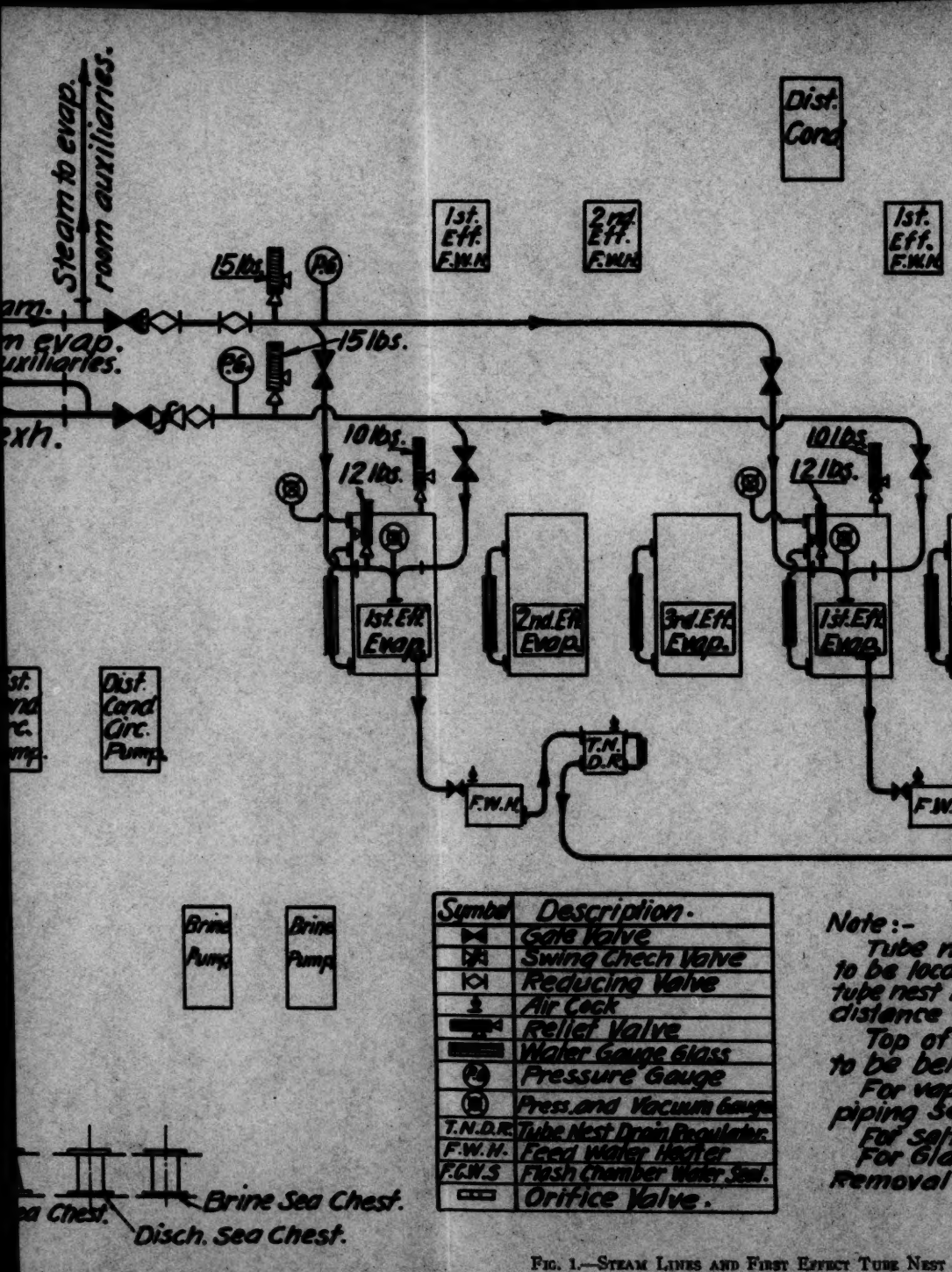
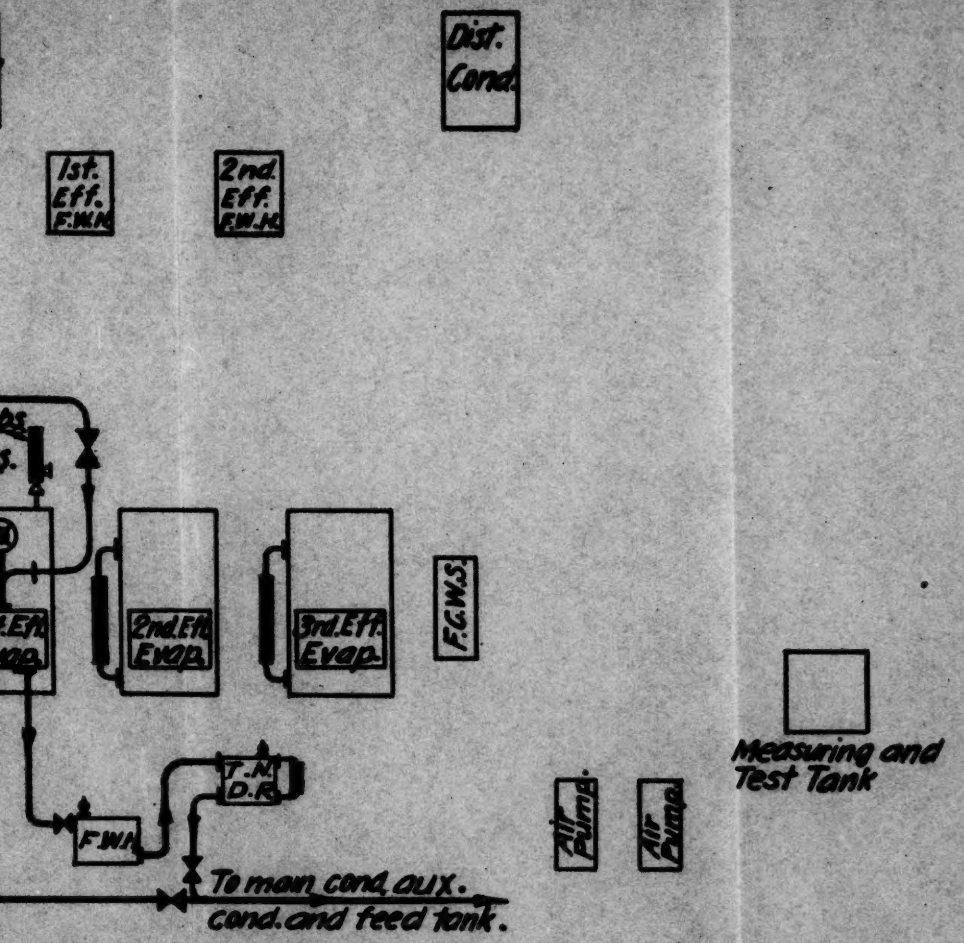


FIG. 1.—STEAM LINES AND FIRST EFFECT TUBE NEST



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Tube nest drain regulators
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Top of feed water heater
o be below regulator.
For vapor and fresh water
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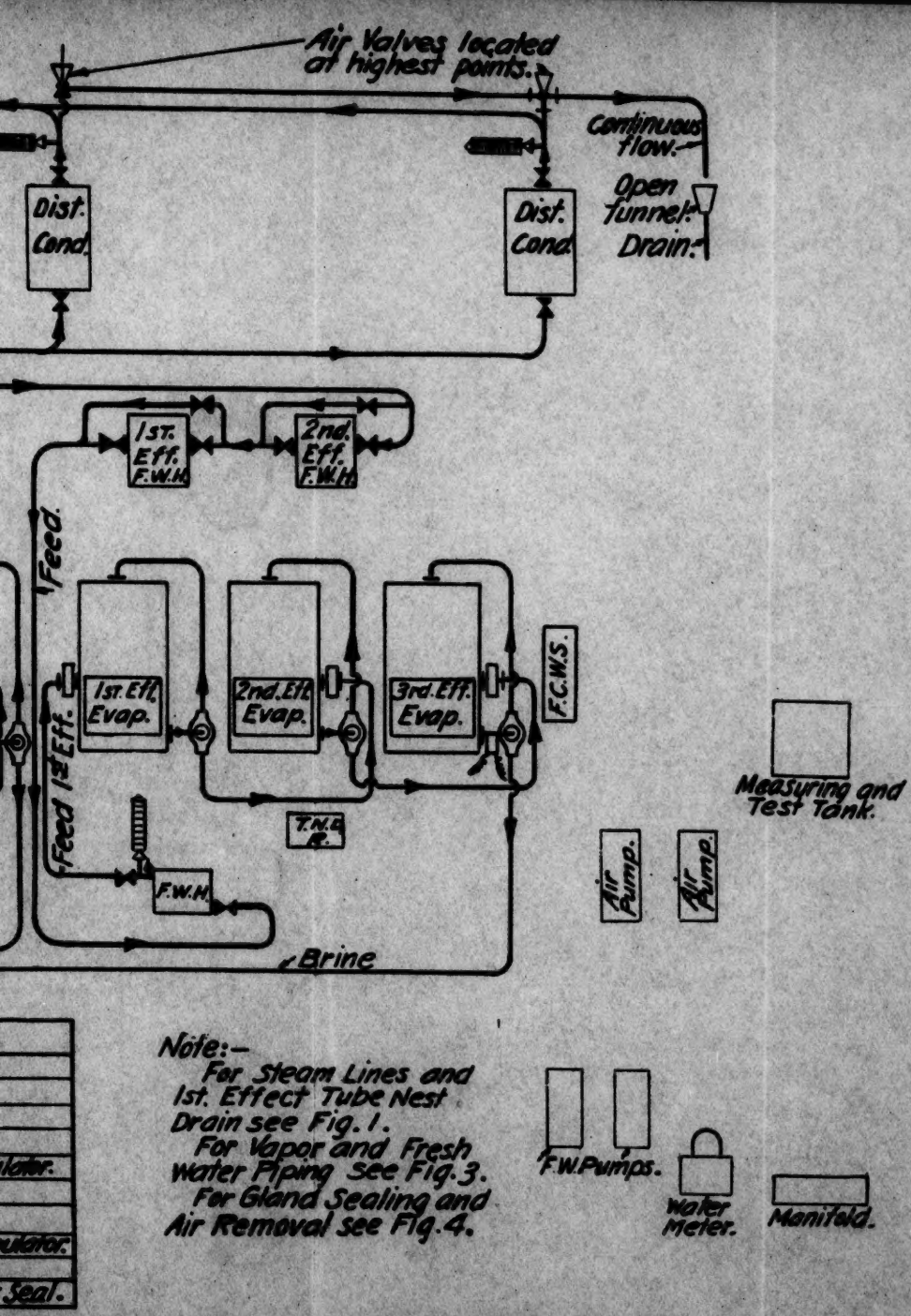
For salt water piping see Fig. 2.
For Gland Sealing and Air
emoval see Fig. 4.

FIG. 2.—SALT WATER PIPING.


Air Valves located at highest points.
Continuous flow.
Open funnel.
Drain.
Dist. Cond.
Dist. Cond.
1st. Eff. F.W.H.
2nd. Eff. F.W.H.
Feed
Feed 1st Eff.
1st. Eff. Evap.
2nd. Eff. Evap.
3rd. Eff. Evap.
F.C.W.S.
T.N.B.
F.W.H.
Brine
Air Pump.
Air Pump.
Measuring and Test Tank.
F.W. Pumps.
Water Meter.
Manifold.
Note:- For Steam Lines and 1st. Effect Tube Nest Drain see Fig. 1. For Vapor and Fresh Water Piping see Fig. 3. For Gland Sealing and Air Removal see Fig. 4.
Seal.

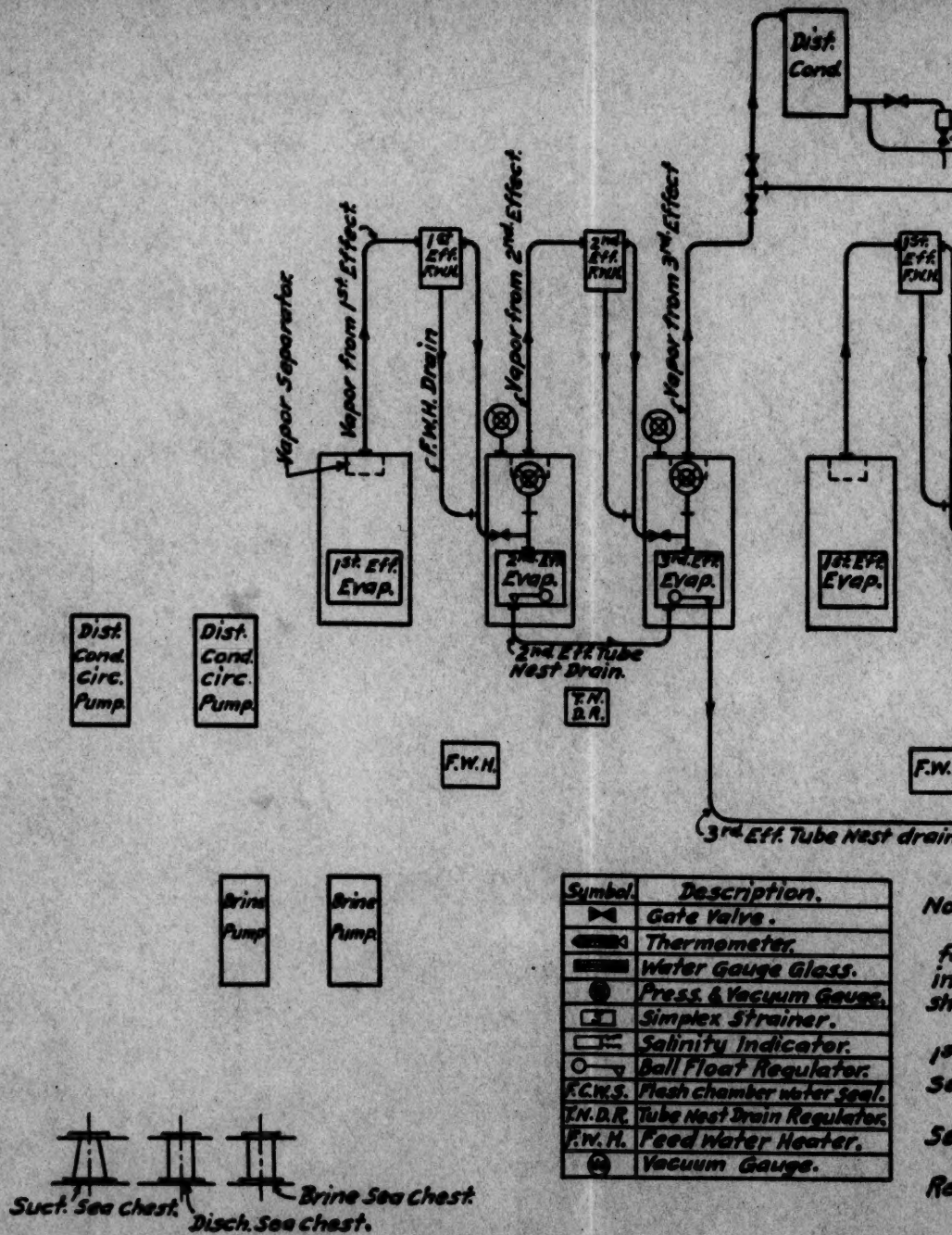
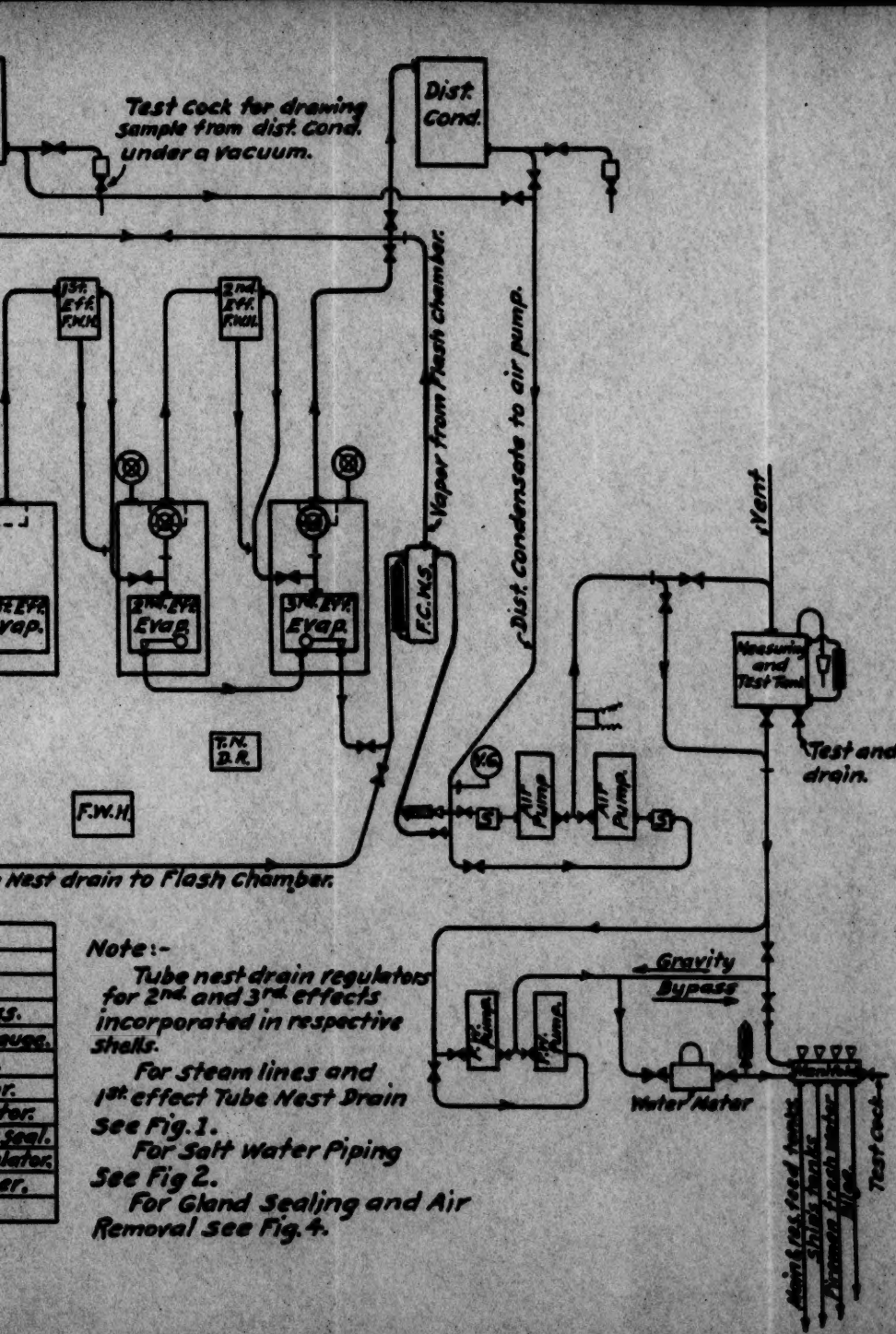


FIG. 3.—VAPOR AND FRESH W



Note:-

Tube nest drain regulators for 2nd. and 3rd. effects incorporated in respective shells.

For steam lines and 1st. effect Tube Nest Drain See Fig. 1.

For Salt Water Piping See Fig 2.

For Gland Sealing and Air Removal See Fig. 4.

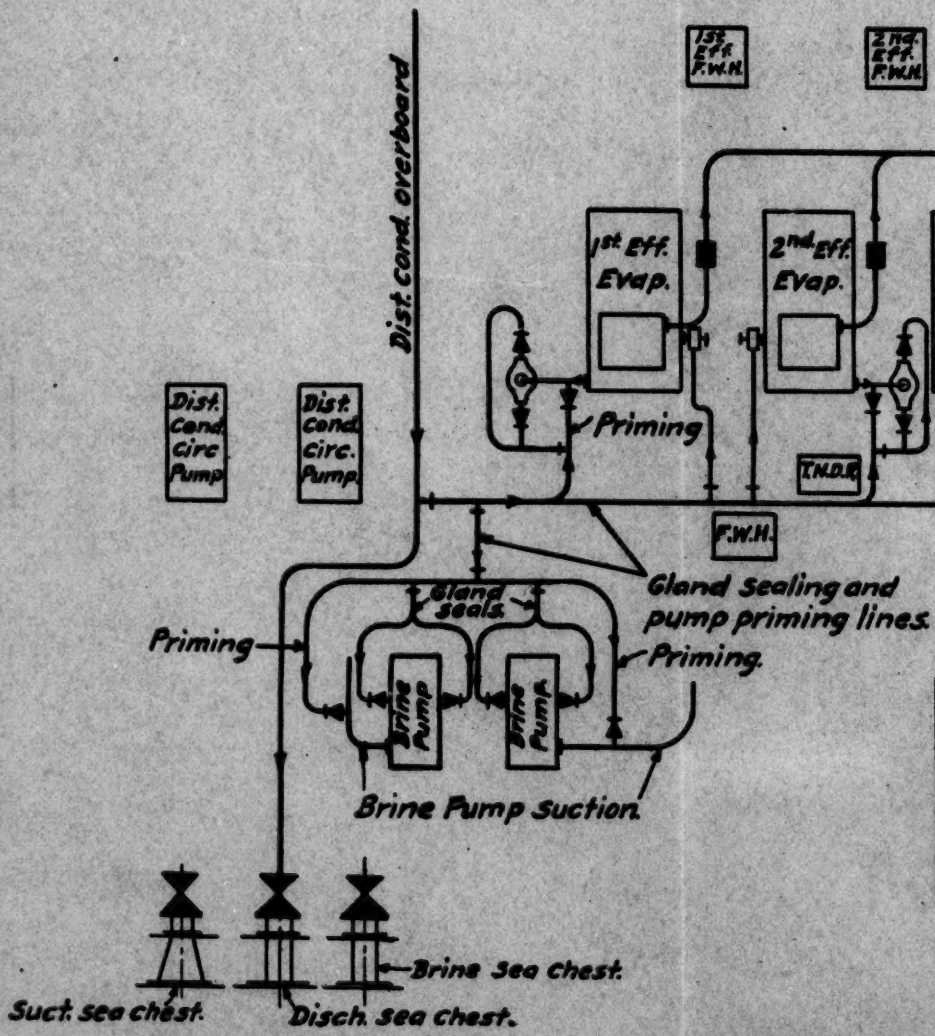
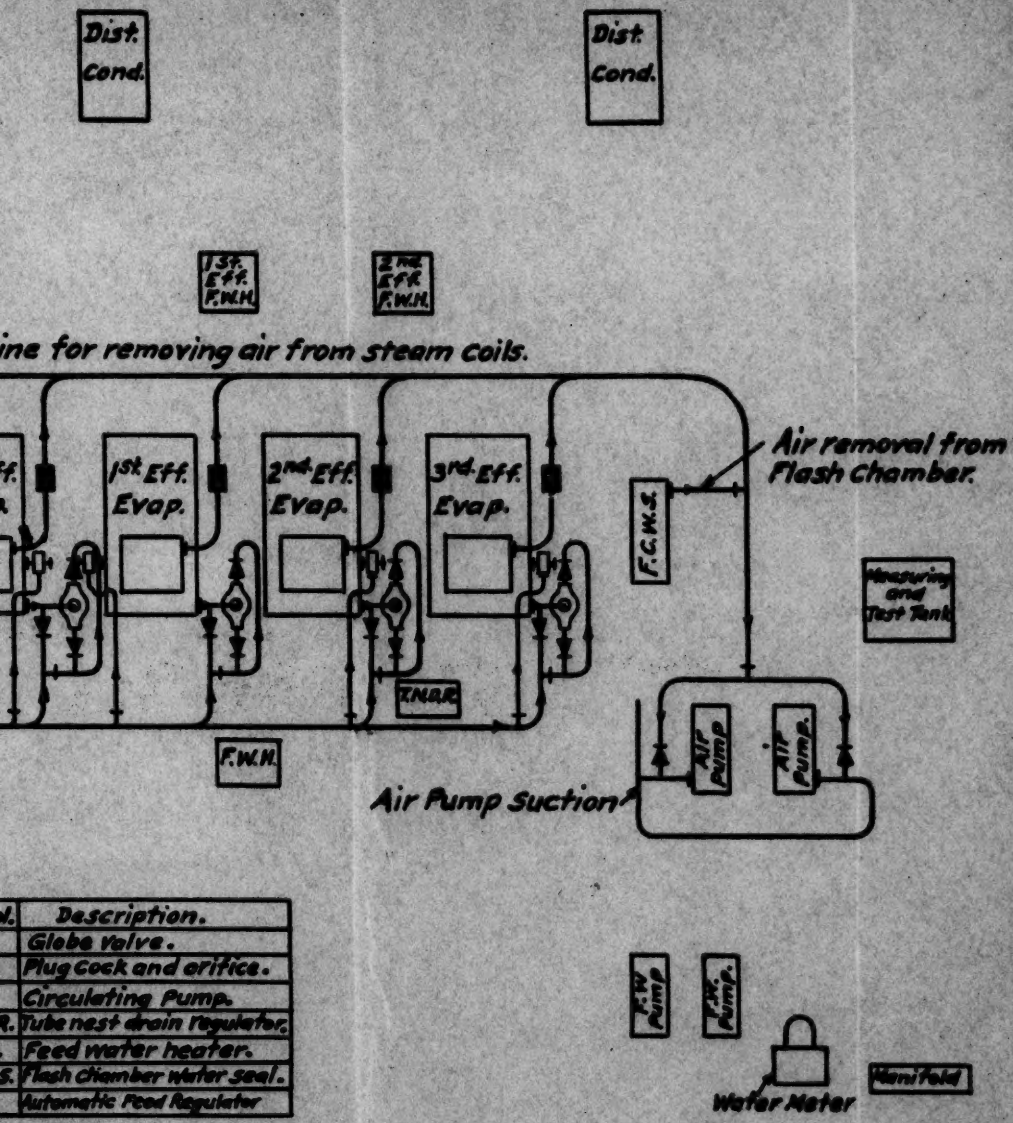


FIG. 4.—GLAND SEALING AND AIR REMOVA



	Description.
	Globe Valve.
	Plug Cock and orifice.
	Circulating Pump.
R.	Tube nest drain Regulator.
	Feed water heater.
S.	Flash chamber Water seal.
	Automatic Feed Regulator

D Air Removal Connections.

all the water a ship needs. In case there is not sufficient auxiliary exhaust, a high pressure trap should be bled into the main feed heater and the steam thus obtained allowed to back up into the auxiliary exhaust line and so find its way into the evaporators. The use of steam in this way obviates the possibility of superheated steam being used in the evaporators, a condition which would obtain in case boiler steam were sent to the evaporators through a reducing valve.

This superheated steam would, of course, have a considerably higher temperature than saturated steam at the same pressure and, as stated before, temperatures over 240 degrees F. should be avoided.

After the steam has gone through the tubes of the first effect coil, it is condensed and returned to the main condenser, auxiliary condenser or feed tank for use again in the boilers. This condensate is, of course, tainted with oil, grease and boiler compound and so is unfit for drinking purposes. To prevent the first effect steam from blowing through the coil, a trap, called the Tube Nest Drain Regulator, is put in the line to the condensers. This regulator maintains a water seal on the first effect tube nest. The condensate from the first effect coil is quite hot, being nearly at the temperature of the entering steam. The heat in this condensate would be lost if it were returned direct to the condensers and would overheat the feed tanks if led direct to them. To take out all possible heat in the condensate, it is run through a heater through which the salt water feed for the evaporators is also passed. This heater is of the U-tube or bow-tube type with about 50 square feet of heating surface. A by-pass is fitted so that the condensate may be taken away around the heater without shutting down the plant. In order to use a feed heater with this condensate, the tube nest drain regulator is put outside the evaporator beyond the feed heater.

A pilot valve, which lifts at 12 pounds, is mounted on the steam line to the tube nest at the evaporator shell. A relief

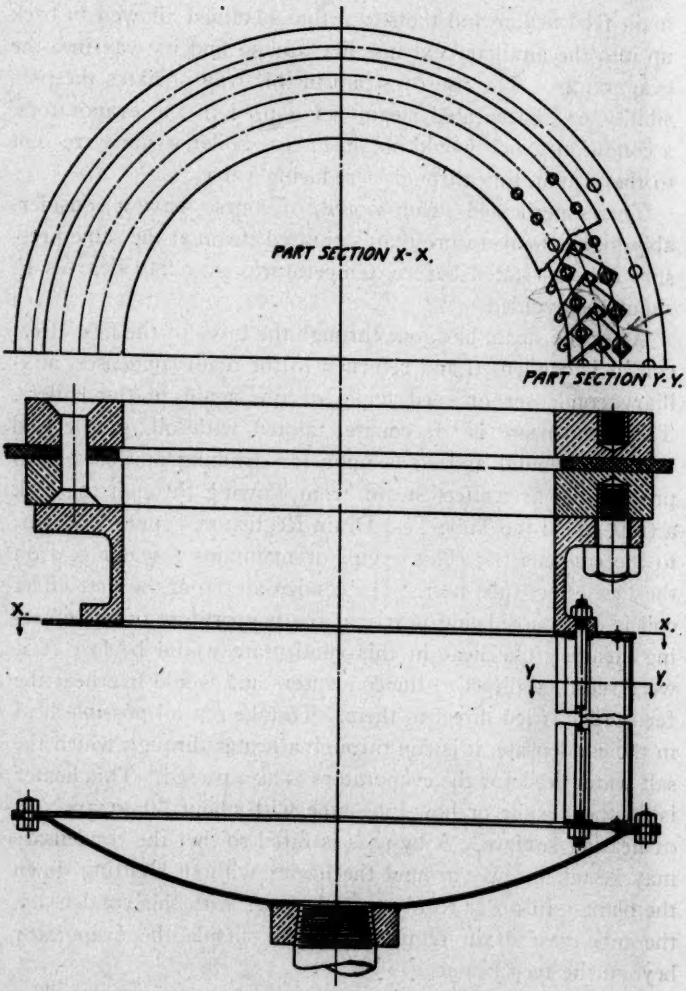


FIG. 3A.—VAPOR SEPARATOR.

valve set at 15 pounds is fitted on the auxiliary exhaust line to take care of surges of pressures in the line due to machinery such as the steering gear.

Great care must be taken to keep the feed heater tight as any leak from the salt water side will salt up the boiler feed water.

The drain regulator operates on the ball float principle, the movement of the ball up and down, opening or closing the valve and so allowing more or less condensed water to flow out of the bottom of the tube nest.

Figure 5a shows the drain regulator and figure 5b shows the feed regulator. It will be noted that the effect of the ball rising or falling produces the opposite effect in the drain regulator from that in the feed regulator. The feed regulator has a priming connection at the valve stem to prevent scale from forming on the stem.

Figure 6 shows the location and connections of feed and tube nest drain regulators.

The tube nest drain regulator, being placed below the tube nest and above the feed heater, insures that the feed heater will always be full of water and consequently that steam will never blow through to the condenser or feed tanks. Great care must be taken to have all the air and equalizing pipes from the regulator absolutely tight.

Figure 7 shows a typical tube nest as fitted in the converted evaporators of the *Texas*. There are in this tube nest, 135 1½-inch O. D. tubes, 4 feet 7 inches long, spaced 2½ inches center to center and giving a heating surface of 240 square feet. The tubes are expanded into the tube sheets at both ends. The front tube sheet is fixed while the after tube sheet is part of a floating head which allows for the expansion of the tubes. Steam enters the top of the steam chamber, is condensed in the tubes and the water drains back through the tubes to the bottom of the steam chamber whence it passes to the main or auxiliary condenser or the feed tanks as noted above.

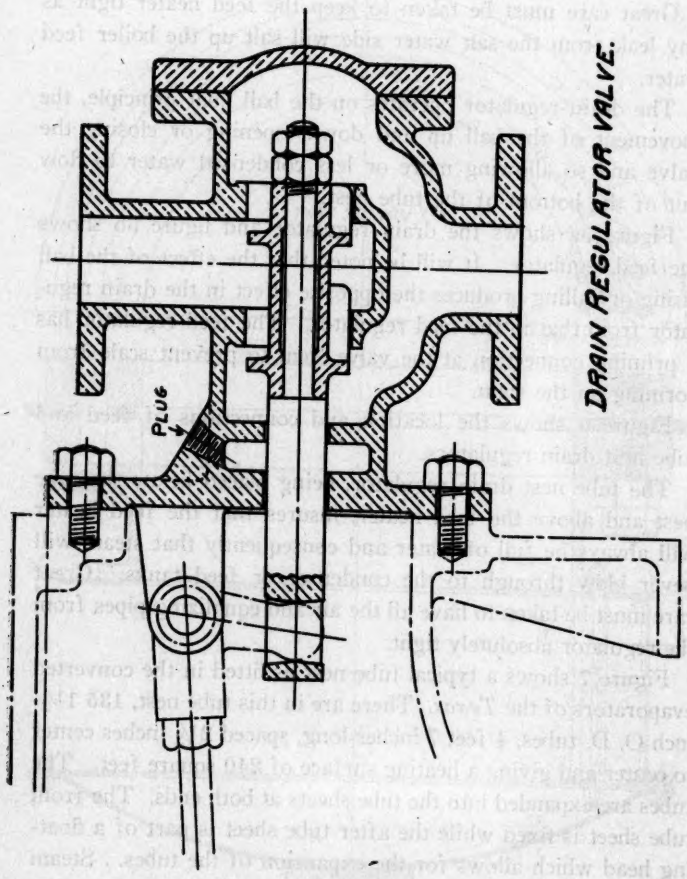


FIG. 5A.—FEED REGULATOR AND DRAIN REGULATOR.

FIG. 5A.—FEED REGULATOR AND DRAIN REGULATOR.

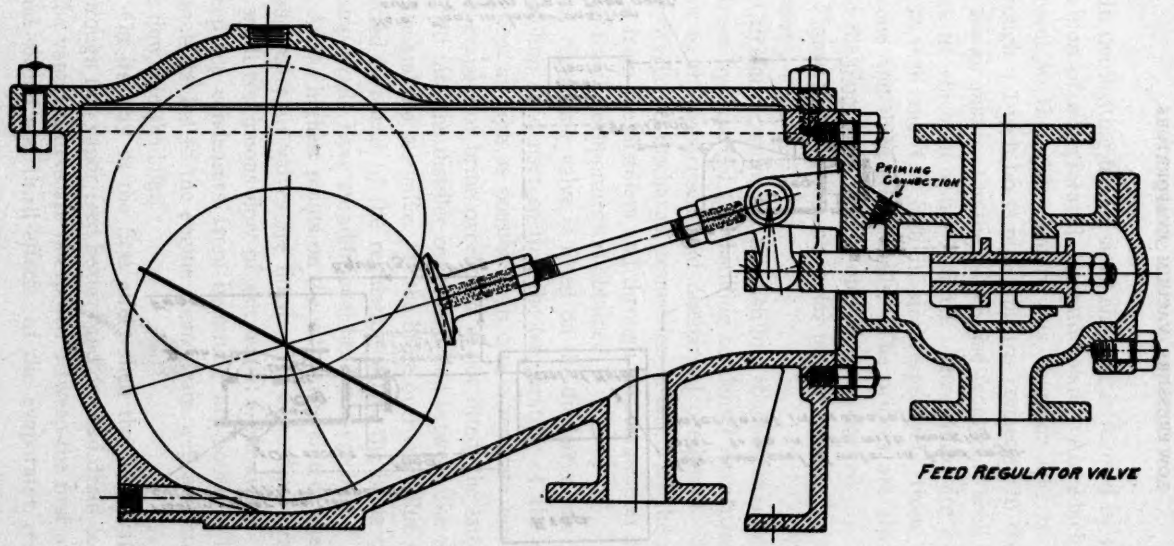


FIG. 5B.—FEED REGULATOR AND DRAIN REGULATOR.

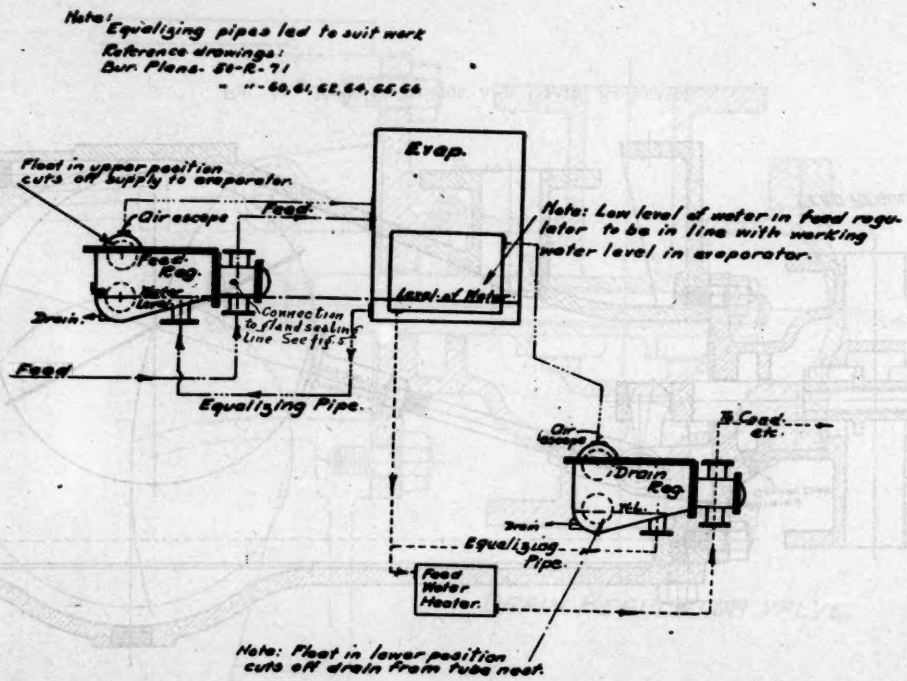

Note: Equalizing pipes led to suit work
Reference drawings:
Bur. Plans- 50-R-71
" " -60, 61, 62, 64, 65, 66
Float in upper position cuts off supply to evaporator.
Air escape
Feed
Evap.
Feed Reg.
Water Level
Drain
Connection to stand sealing line See fig 5
Level of Water
Note: Low level of water in feed regulator to be in line with working water level in evaporator.
Feed
Equalizing Pipe.
Air escape
To Feed etc.
Drain Reg.
Drain
Equalizing Pipe.
Feed Water Heater.
Note: Float in lower position cuts off drain from tube nest.

FIG. 6.—LOCATION AND CONNECTIONS OF FEED AND TUBE NEST DRAIN REGULATORS.

In the floating head a small chamber is formed by the interposition of a perforated separator plate. Air which may be present in the steam enters this chamber and is removed through a 1-inch brass pipe which is connected to the suction of the air pump. On the 1-inch line from each shell is a plug cock in which an orifice is fitted. The orifices are 1/16 inch, 3/32 inch and 1/8 inch in size respectively from the first, second and third effect shells. The orifices are always open and the plug cocks may be opened when desired.

Figure 2 shows the salt water piping systems for the evaporators.

Circulating water for the distilling condensers is taken from the sea by either of the distilling condenser circulating pumps, sent to the condensers and discharged overboard. Inside of the overboard discharge, connection is taken off the discharge line from the distillers and through this connection salt water feed for the evaporators is obtained.

A regulating valve is fitted on the discharge line from the distilling condenser so that the temperature of the feed may be kept as high as possible. In case a separate sea chest for the evaporator brine overboard is not fitted, the latter should be led into the distiller overboard line between the regulating valve and the sea valve. If the brine overboard line were connected in ahead of the regulating valve, the brine overboard pump would have an additional head to overcome.

At the highest points on the salt water side of the distilling condensers air valves are fitted. These are kept cracked and a small continuous flow of salt water is led away. This will keep the condensers from becoming air bound. The drain should be led to the engine room drain well and not allowed to flow into the bilge.

On its way to the first effect shell the feed water passes through two vapor feed heaters and one coil drain feed heater. The vapor heaters are interposed between the first and second and second and third effects, of the evaporator respectively.

In a quadruple plant a vapor heater is also put between the third and fourth effects. In some plants there is insufficient room to put the vapor heaters in the vapor lines. In these cases the heaters are placed in the steam heads of the second and third (and fourth) effects. The results are identical. The vapor from the first effect shell goes through the first effect heater on its way to the coil of the second effect. Likewise the vapor from the second effect shell goes through the second effect heater on its way to the coil of the third effect. After going through the vapor heaters, the feed goes through the first effect coil drain heater where its temperature is further raised. The economy obtained by thus using the vapor and first effect coil drains for raising the feed temperature is very considerable as will be shown later on by concrete examples.

Figures 8 and 9A and 9B show the types of vapor feed heaters employed. By-passes are fitted on the salt water side of these heaters so that a salt water leak will not necessitate a shut down of the plant.

After going through the heaters, the feed passes to the shell of the first effect evaporator through a feed regulator which maintains the feed water level just above the bottom row of tubes. This is a great departure from evaporators used heretofore.

From the bottom of the shell a centrifugal brine circulating pump takes its suction and discharges part of the water to the next effect shell and the other part to the top of its own shell. This latter part is discharged through a nozzle upon a perforated distributing plate which is mounted slightly above the tube nest and completely covers the top of it.

The height of the plate above the tube nest is quite important; if put too close to the tubes the vapor pressure of the evaporated water may prevent water from flowing through the perforations of the plate, thus preventing circulation over the tubes.

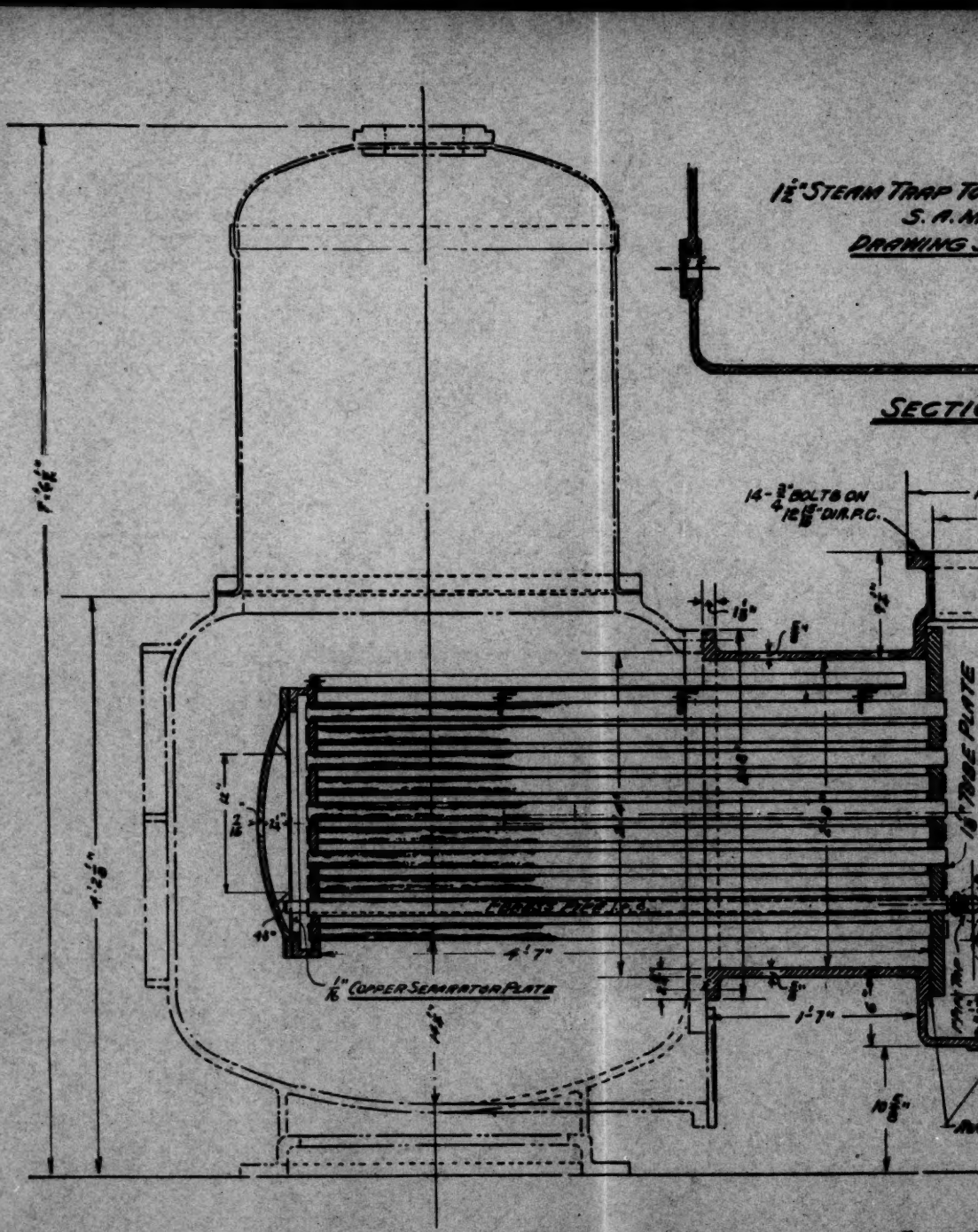

1½" STEAM TRAP TO
S. A. M.
DRAWING
SECTI
14-¾" BOLTS ON
12⅝" DIA. P.C.
18"
8"
2½"
TUBE PLATE
4'-7"
1'-7"
1/16" COPPER SEPARATOR PLATE
45°
10⅝"

FIG. 7.—

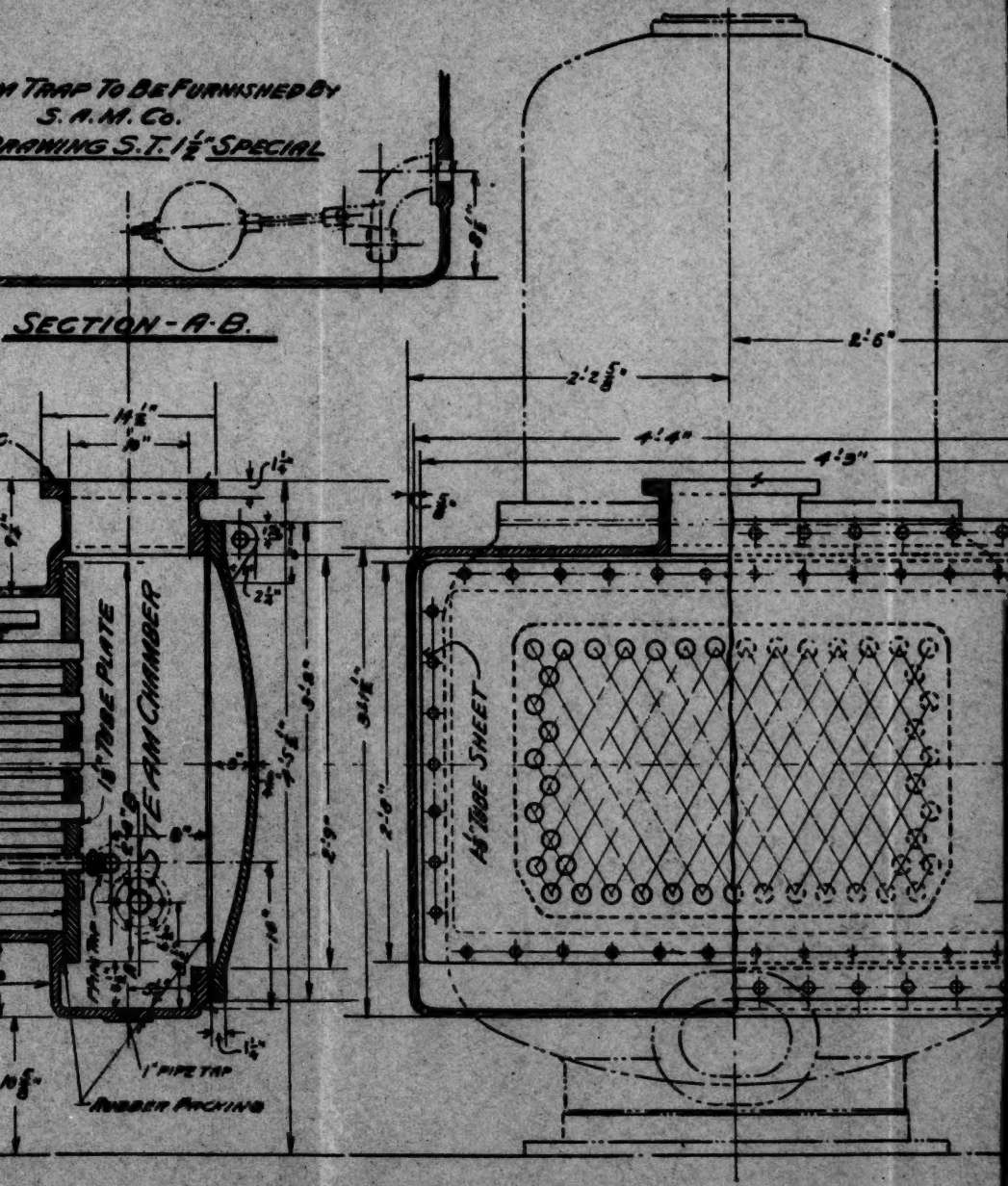


FIG. 7.—EVAPORATOR TUBE NEST.

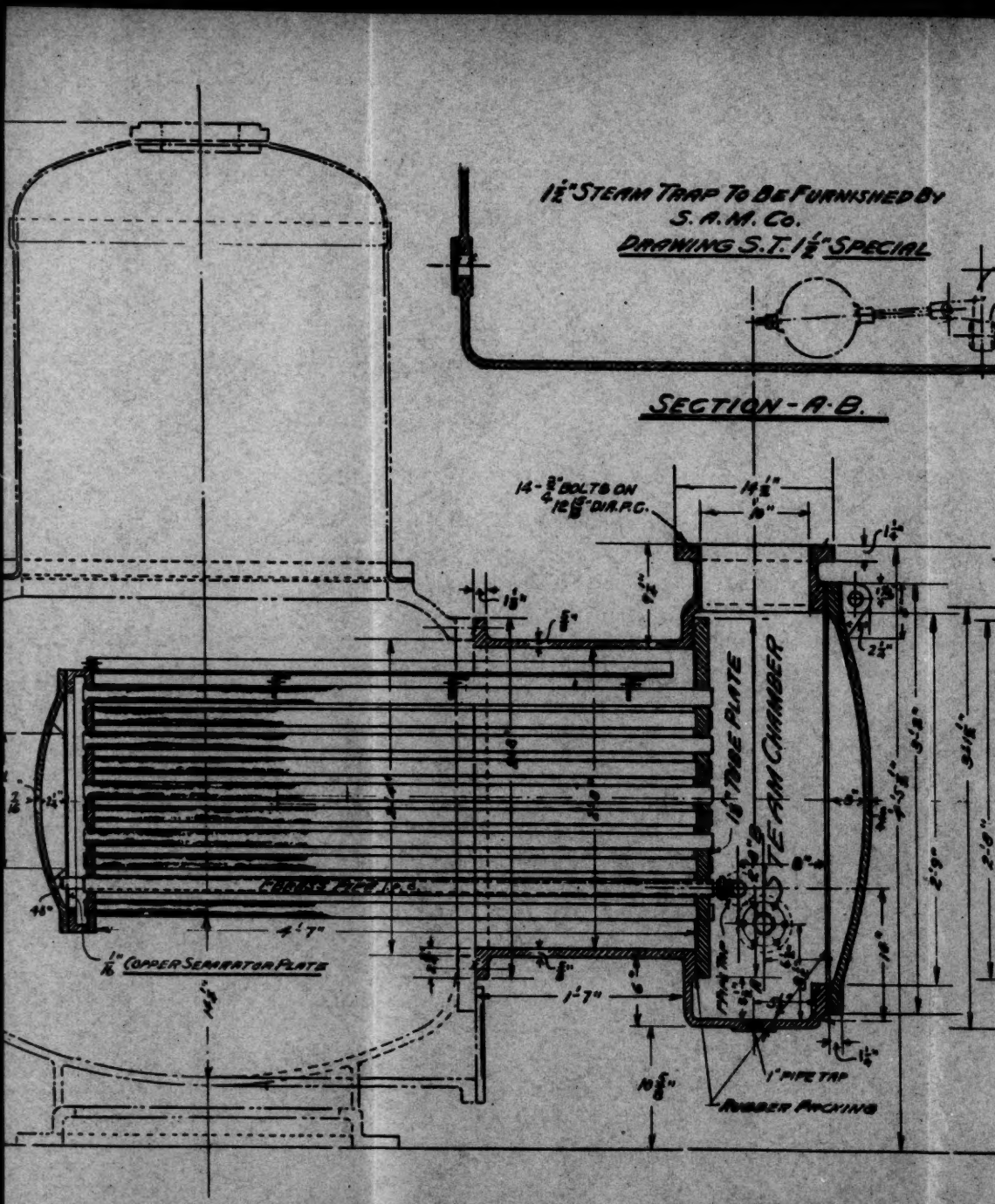
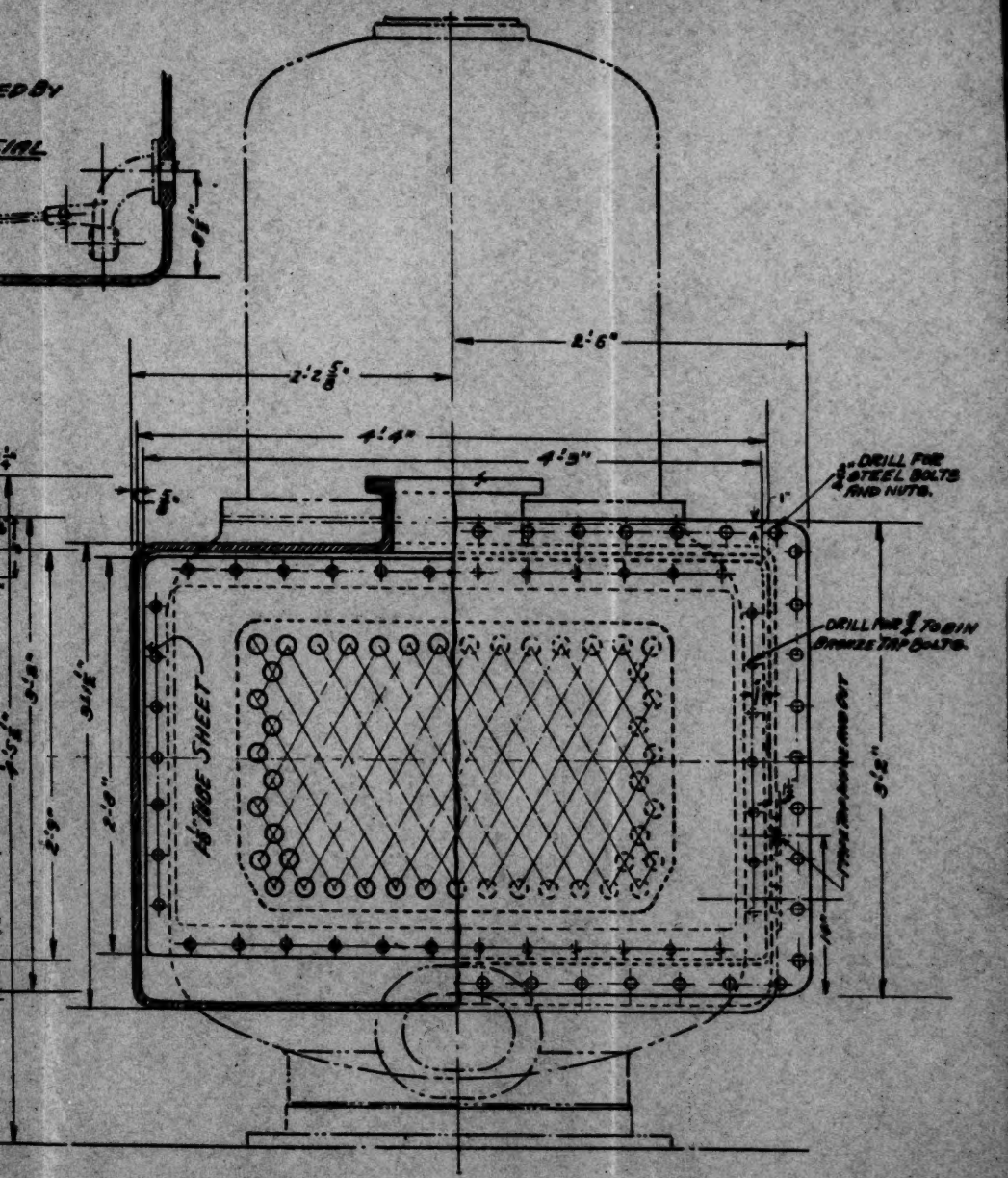
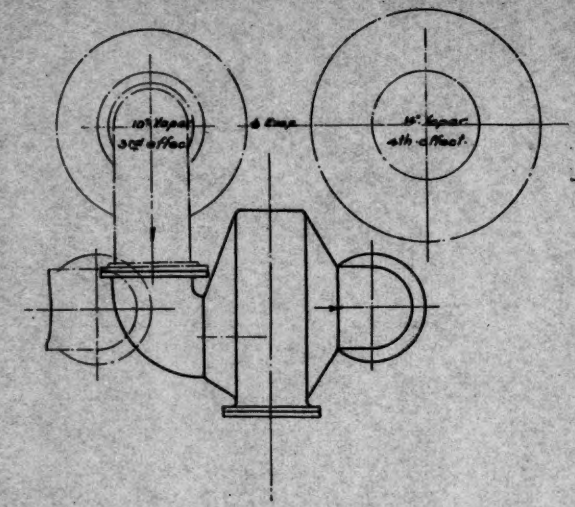
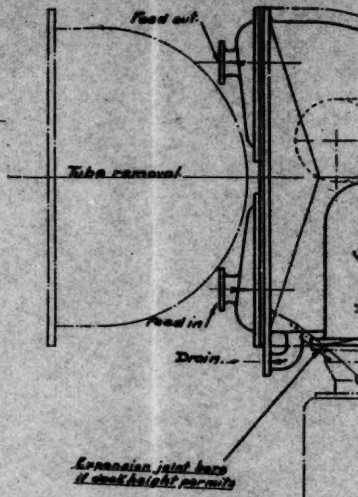


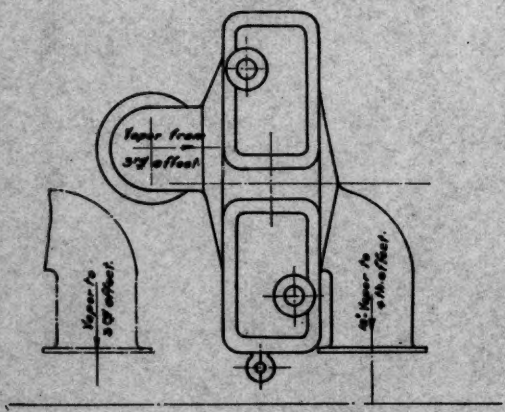
FIG. 7.—EVAPORATOR TUBE NEST.

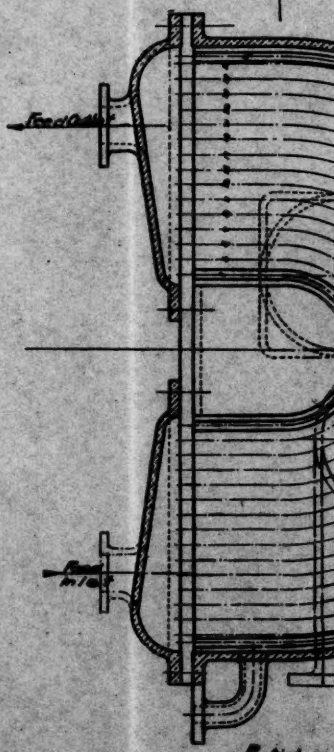

2'-6"
2'-2 5/8"
4'-4"
4'-3"
DRILL FOR
STEEL BOLTS
AND NUTS.
DRILL FOR
BRONZE TAP BOLTS.
AS TUBE SHEET
3'-2"

NEST.









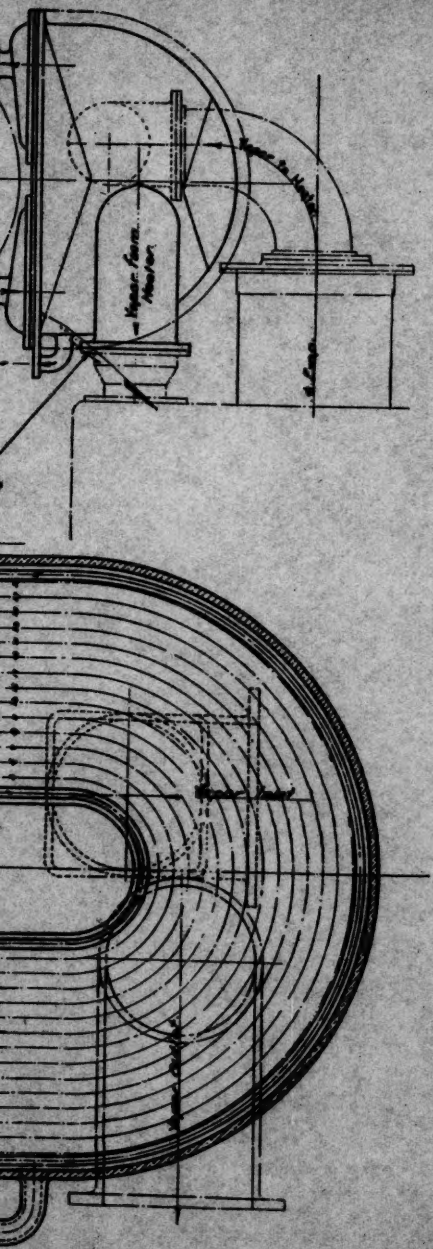
Arrangement of Heater
1½" = 1'

Heater data:-
Surface - 62 sq. ft.
Tubes - 30 - ¾": 0.0-.028 mils thick.
Three pass

Note:
Provision made for lagging, securing, air removal and baffling.

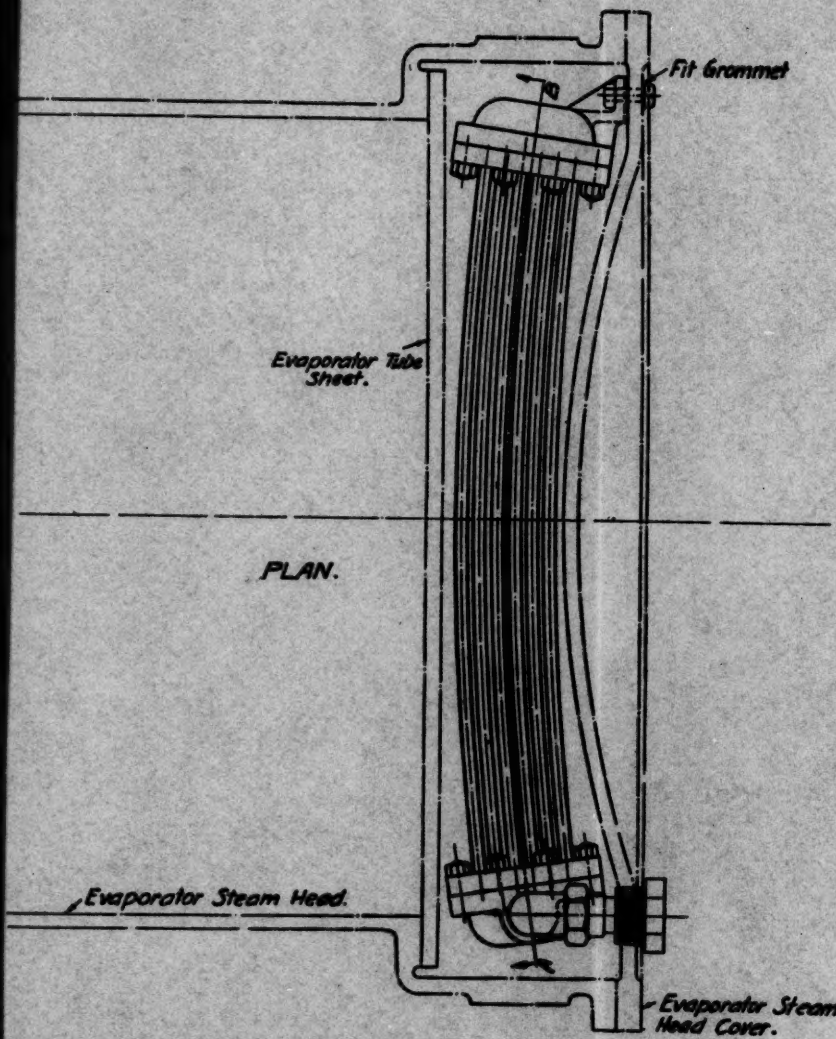
Partial ele

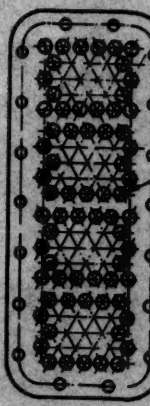
FIG. 8.—VAPOR FEED HEATER (VAPOR LINE TYPE).



Partial section on 'AB'.

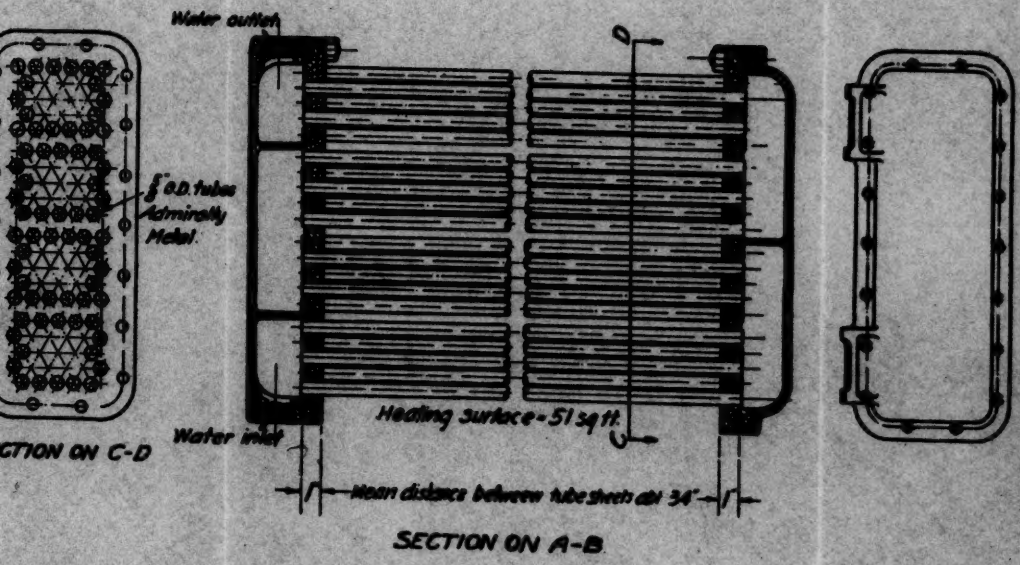
nt).



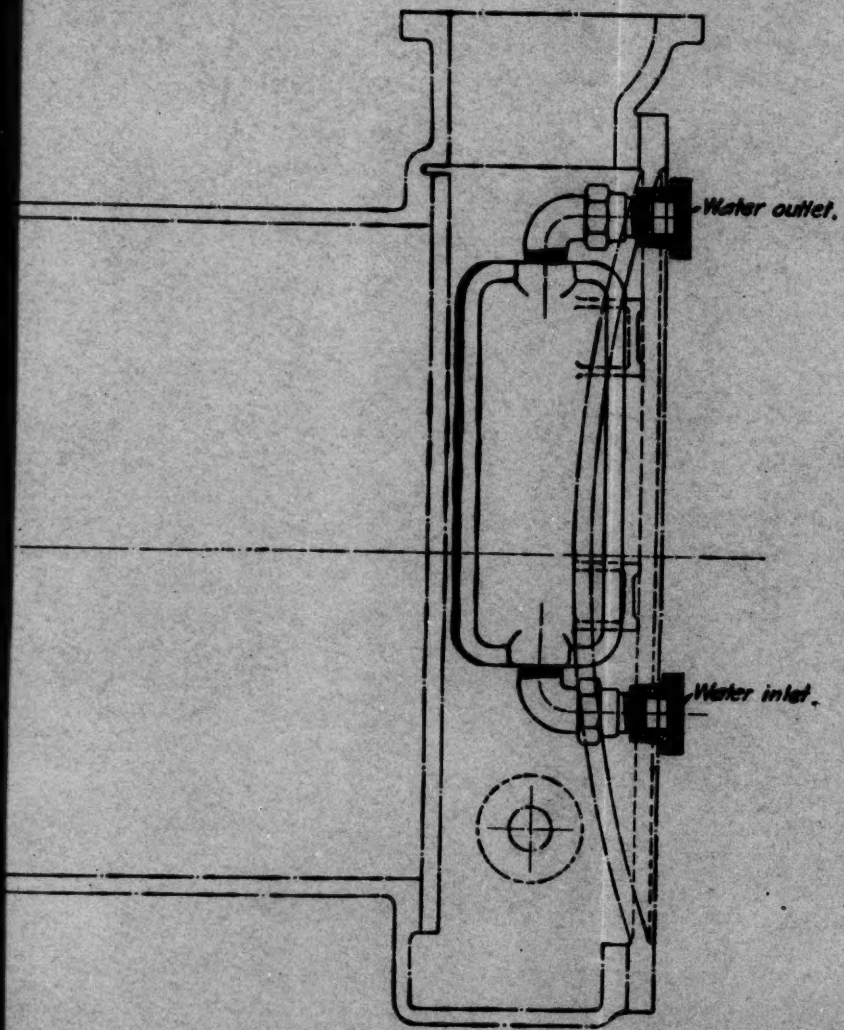


SECTION ON C

FIG. 9A.—VAPOR FEED HEATER (STEAM HEAD


Water outlet
⅝" O.D. tubes Admiralty Metal.
CTION ON C-D
Water inlet
Heating surface - 513 sq ft.
Mean distance between tube sheets abt 34"
SECTION ON A-B

(Steam Head Type).



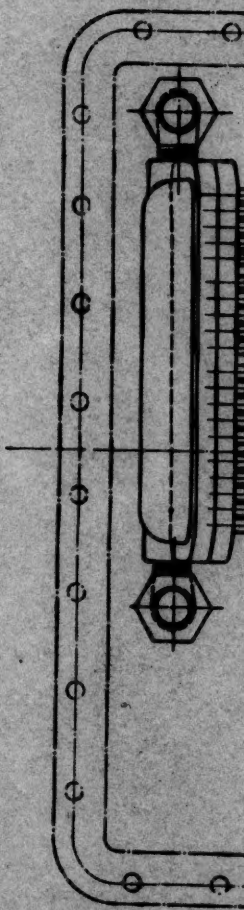
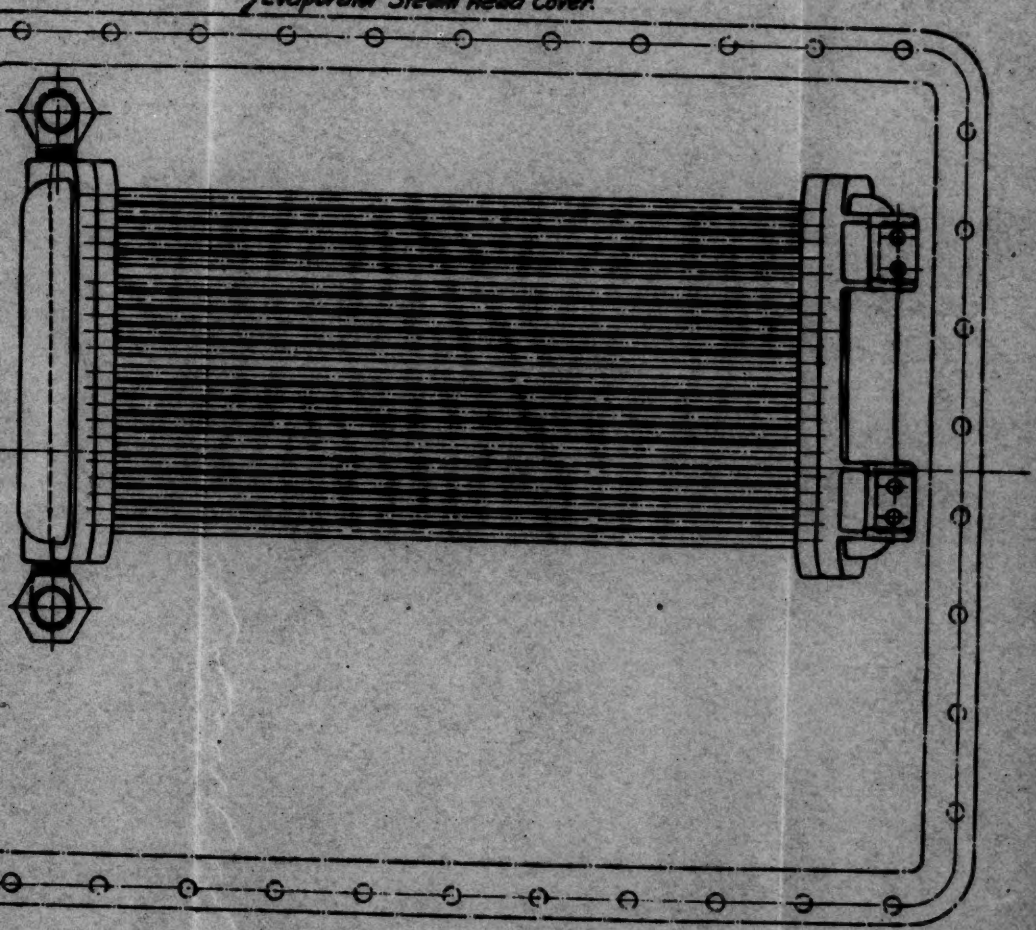


FIG. 9B.—VAPOR FEED HEATER (STEAM HEAD



STEAM HEAD TYPE).

Various forms of distributing plates have been experimented with in the endeavor to get better circulation and hence more evaporation. One type consisted of a box manifold into which the circulating pump discharged and from which slotted pipes extended over the tube nest. The distribution of feed with this type was very good but the vapor rising between the pipes entrained so much water that the salinity of the evaporated water was too high.

It is necessary to have a distributor which will prevent the vapor generated on the tubes from rising directly up to the top of the shell. By forcing the vapor out at the sides of the tube nest, it must change direction to get to the top of the shell. In other words the distributor must act as a baffle.

Another type of distributor tried was one made in the form of a covered box into which the circulating pump discharged. It was desired to keep this box full and under pressure thus insuring a positive head on all the distributing holes in the bottom of the box. It was found, however, that the box could not be kept full unless the size of the holes was materially reduced. This was objectionable due to the liability of the small holes filling up gradually from scale deposits.

The type of plate which will be installed is a flat copper plate with raised edges. The plate should not be thicker than 1/8 inch and the holes should not be less than 3/8 inch in diameter.

The spacing of holes should be such that the streams of water will come down both upon and between the tubes in the upper row. The spacing in the other direction should be close enough so that water will find its way to all parts of the tubes as nearly as possible.

Enough circulating water must be sent over the plate to insure a flow through all the holes without enough excess to allow a spattering from the top of the plate as this excess will be picked up by the escaping vapor.

Figure 10 shows the type decided upon as a standard.

As the salt water descends through the tube nest, part of each stream is evaporated. The remainder of the stream partially washes off such scale or deposit as may be left from the evaporated water. To examine the tubes for scale and to determine if the circulation is satisfactory, sight glasses are fitted in each side of the evaporator shell. These glasses will be about 10 inches in diameter and fitted so that the frames may be removed from the shell. The advantage of such a method of observation is obvious.

Some scale does form on the tubes but after a certain thickness has formed, it will crack off and drop to the bottom of the shell. If these pieces of scale were allowed to get into the brine circulating pump they would clog it up and if discharged back to the distributing plate would choke up the holes. To prevent this clogging by scale a sump tank is fitted at the bottom of each evaporator shell.

Figure 11 shows the arrangement of the sump tank. It is seen that scale which cracks off the tubes falls to the bottom of the sump from which it can be removed through a cleaning door. A horizontal screen prevents any scale being carried to the pump by the velocity of the circulating water. The screen must be removed and cleaned frequently and the scale should be removed from the sump at the same time.

That part of the salt water not discharged back into the first effect shell goes to the second effect shell through a regulator where it, in turn, is partly circulated and partly sent on to the third effect.

In the third effect that part of the water not circulated is discharged to the suction of the brine overboard pump which sends it to the sea.

The amount of water taken into the system depends on the concentration of the brine in the last effect and on the amount evaporated, the feed regulators maintaining a constant level in each of the evaporator shells. The valve on the line leading

MUNTZ METAL
BOLTS

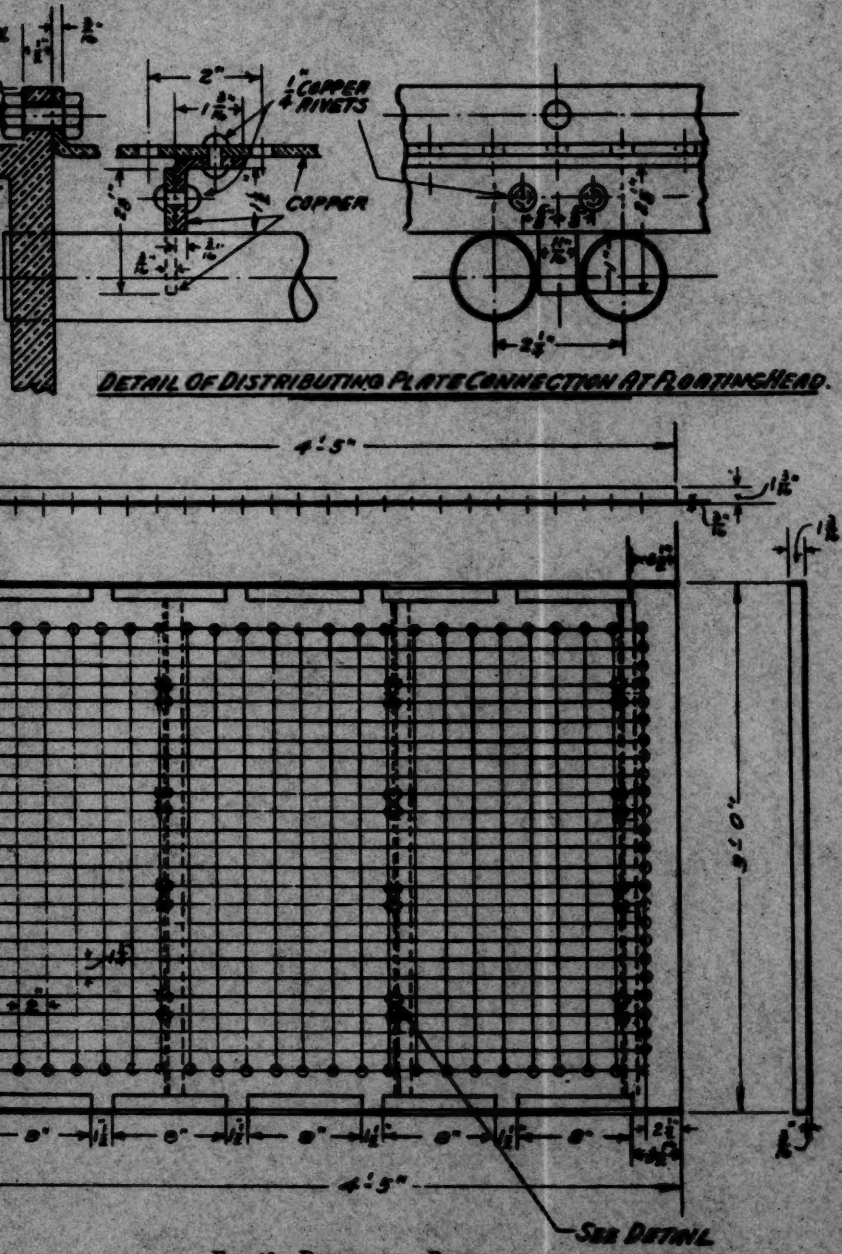
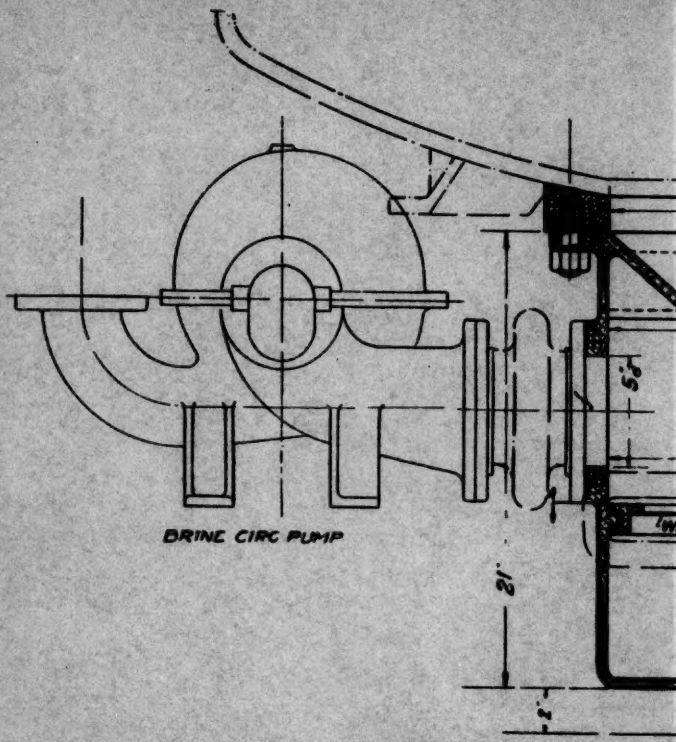


FIG. 10.—DISTRIBUTING PLATE.



SECT

Fig. 11.

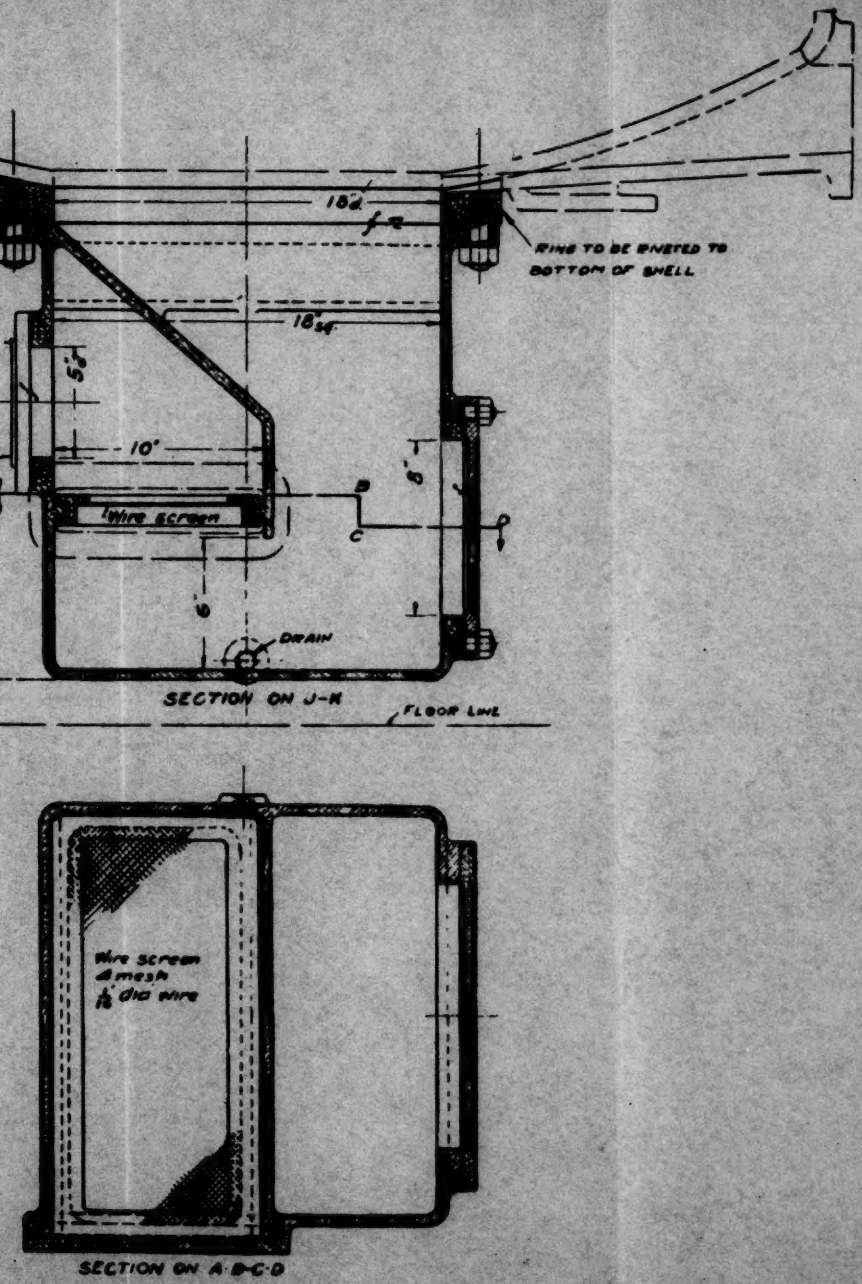


FIG. 11.—SUMP TANK.

to the brine overboard pump should be regulated so that the concentration of the brine going overboard is kept at about 2½/32.

In the feed line to the second and third effects and in the discharge from the third effect an electric concentration salinity indicator cell is fitted. Leads from each cell are taken to an indicator on which can be read off the concentration of the brine in each effect.

Figure 3 shows the vapor and fresh water piping systems.

The steam made in the first effect shell rises to the top of the shell and passes through a vapor separator to the vapor pipe. The vapor separator is shown in Figure 3-a. Steam enters all around the periphery of the separator, the entrained water being removed by the hook-shaped baffles and draining back into the shell through the drain pipe.

On its way to the steam head of the second effect, the vapor passes through a heater where part of the vapor is condensed giving up its latent heat to the feed water passing through the water side of the heater.

The remaining vapor and condensate pass together into the steam coil of the second effect where the steam is condensed in giving up its latent heat to the water in the second effect shell. The condensate from the second effect coil passes to the steam head of the third effect coil where it gives up some of its remaining heat.

The second effect vapor passes through a separator and vapor heater to the steam coil of the third effect where it is condensed giving up its heat to vaporize the water in the shell.

The condensate from the third effect shell passes to a flash chamber where part of it flashes into vapor which goes to the distiller. The remainder goes to the suction of the air pump. The flash chamber is fitted so that it forms a water seal preventing any vapor going to the air pump.

The vapor from the third effect shell joins with the vapor from the flash chamber and goes to the distiller where it is

condensed. The condensate is taken by the air pump and discharged to the measuring and test tank. In the discharge from the air pump is fitted a distillate salinity cell from which leads are taken to the salinity indicator where the salinity can be measured. Connections are also made to automatic signal lights which show at all times the salinity of the distillate. A white light shows when the distillate has less than .3 of a grain of chlorine per gallon, a green light shows when the distillate has from .3 to 5.0 grains per gallon while a red light comes on when the concentration is over 5.0 grains per gallon. These indications roughly show the limits of water, (*a*) good for the boilers, (*b*) fit for drinking, but not for the boilers, and (*c*) unfit for use at all, respectively. From the measuring tank the water flows by gravity through a meter to the reserve feed bottoms or to the fresh water pumps, which discharge it through the meter and a manifold to feed tanks, ship's tanks or firemen's wash room tanks.

The auxiliaries for the converted plants will all be motor driven. The distiller circulating, brine circulating, brine overboard and fresh water pumps are single stage centrifugal pumps direct connected to the motors. The air pumps are reciprocating and connected to the motors by spur or worm gearing. In connecting up and using the centrifugal pumps, care must be taken to keep the glands sealed at all times. If not so sealed, the vacuum will be broken and the glands will overheat and burn out. Care must be taken also to keep the pumps from getting air or vapor bound. If the suction of the pump is not kept flooded, the pump chamber will fill with air or vapor and the pump will refuse to pump. In case this happens, the pump should be primed at once. In installing priming connections they should be arranged as follows: They should be as high up in the suction as possible; they should be as nearly parallel to the suction as possible, and they should be as close to the pump as possible. This will insure that the priming water will be directed into the suction chamber.

Figure 4 shows the connections for sealing the glands of the pumps and keeping their suctions flooded.

The distilling condensers are built along the same lines as the main and auxiliary condensers.

The effect of feed heating on the capacity of the evaporating plant is shown by the following examples. Figure 12 shows a diagrammatic layout of the plants. In each case the problem is based on 1 pound of steam supplied the evaporator at 25 pounds pressure absolute (10-pound gauge), feed water temperature coming from distilling condensers at 90 degrees F. and a vacuum on the third effect shell of 26 inches. In each case the concentration of the brine overboard is to be kept at 2½/32ds. It is assumed that there are equal drops in pressure from one effect to the next. This has been found approximately true from test results; that is,

	Pressure Absolute	Temperature Degrees F.
1st effect coil.....	25	240
1st effect shell.....	17 1/3	220
2d effect shell.....	9 2/3	192
3d effect shell.....	2	126

We will assume also that in each heater, the steam and feed water or condensate and feed water temperatures will be brought within 20 degrees of each other. The temperatures given in Fig. 12 are deduced by this method.

EXAMPLE 1. VAPOR HEATERS AND COIL DRAIN HEATER.

Let us assume that 4 pounds of feed water is used for each pound of steam.

The entering steam is at 240 degrees and it will leave the first effect coil at the same temperature so that the heat given up for evaporation is the latent heat at 240 degrees or 952 B.T.U.

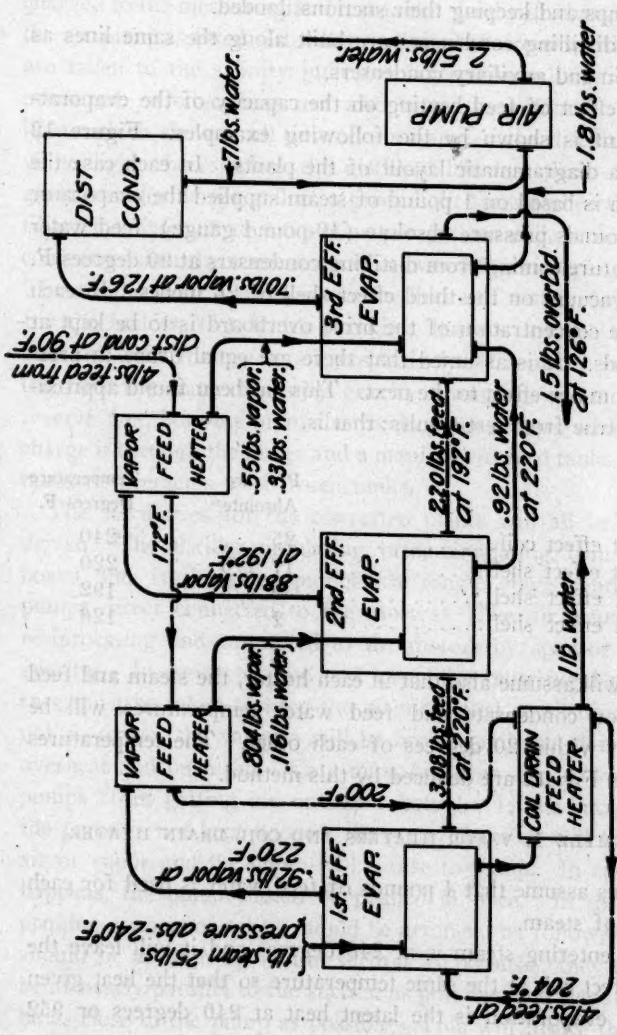


Fig. 12.—Example 1. Triple effect evaporator with vapor heaters and coil drain heater.

FIG. 12.—VALUE OF FEED HEATING.

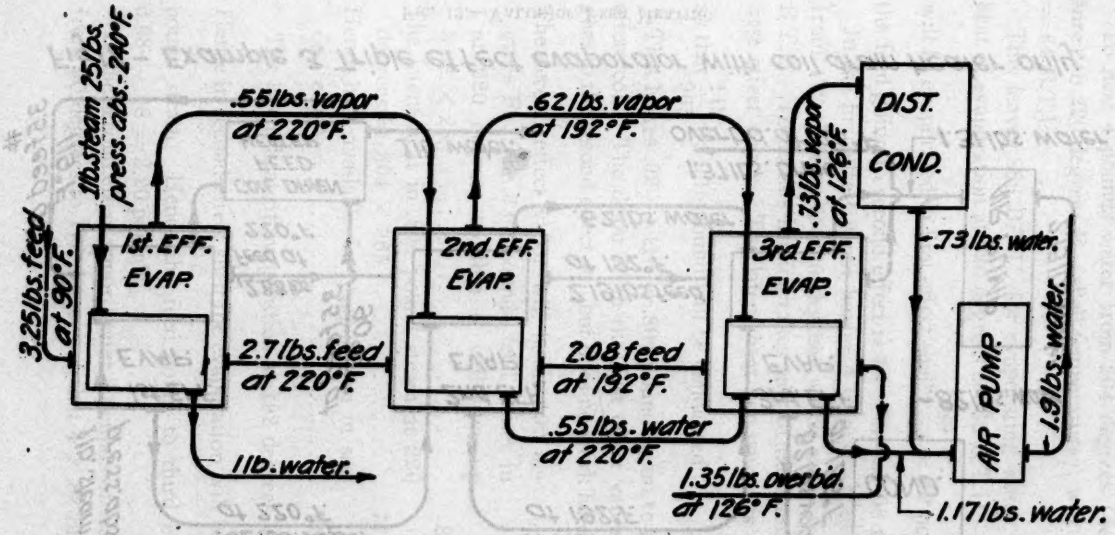


Fig.12-Example 2. Triple effect evaporator with no feed heating.

FIG. 12.—VALUE OF FEED HEATING.

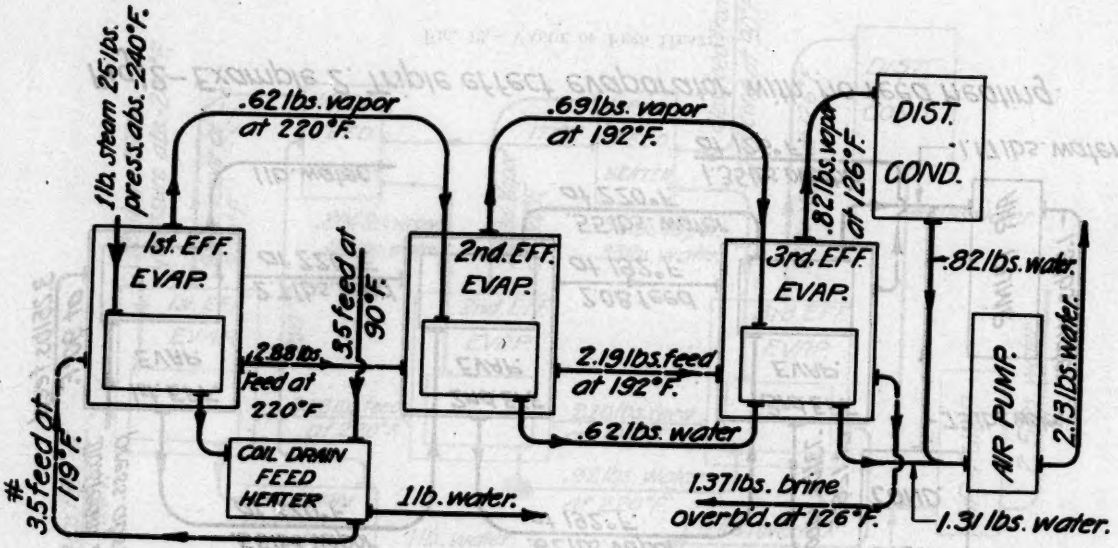

1 lb. steam 25 lbs. press. abs. -240°F.
.62 lbs. vapor at 220°F.
.69 lbs. vapor at 192°F.
.82 lbs. vapor at 126°F.
DIST. COND.
1st. EFF. EVAP.
2nd. EFF. EVAP.
3rd. EFF. EVAP.
.82 lbs. water.
.35 feed at 90°F.
2.88 lbs. feed at 220°F
2.19 lbs. feed at 192°F.
AIR PUMP.
2.13 lbs. water.
.35 feed at 119°F.
COIL DRAIN FEED HEATER
1 lb. water.
.62 lbs. water
1.37 lbs. brine overbd. at 126°F.
1.31 lbs. water.

Fig. 12 – Example 3. Triple effect evaporator with coil drain heater only.

FIG. 12.—VALUE OF FEED HEATING.

To heat 4 pounds water from 204 degrees (feed temperature) to 220 degrees F. (shell temperature), will take $4 \times 16 = 64$ B.T.U.

This leaves 888 B.T.U. for evaporating water. The latent heat of evaporation at 220 degrees is 965 B.T.U. Hence there will be evaporated in first effect shell $\frac{888}{965} = .92$ pound. Hence the feed going to second effect is $4 - .92 = 3.08$ pounds.

In the first effect vapor heater 4 pounds of water is heated from 172 degrees to 200 degrees which takes $4 \times 28 = 112$ B.T.U. To do this heating takes part of the latent heat in the first effect vapor. The latent heat per pound is 965 B.T.U., so that $\frac{112}{965} = .116$ pound will be condensed.

This leaves $.92 - .116 = .804$ pound steam to go to second effect coil. The drain from the first effect vapor heater also goes to the second effect coil but does no work here.

The second effect vapor will be at 192 degrees F. The 3.08 pounds feed comes in at 220 degrees so that it will give up $3.08 \times 28 =$	86.24 B.T.U.
The latent heat in .804 pound steam at 220 degrees $= .804 \times 965 =$	775.86 B.T.U.
Hence heat available for evaporation in second effect = ...	862.10 B.T.U.

The latent heat of evaporation at 192 degrees is 983 B.T.U. Hence in second effect shell the evaporation will be $\frac{862.1}{983} = .88$ pound steam. Hence the feed going to third effect will be $3.08 - .88 = 2.20$ pounds.

In the second effect vapor heater 4 pounds of water is heated from 90 degrees to 172 degrees so that 4×82 or 328 B.T.U.

will be needed. This will be taken from the second effect vapor, which at 192 degrees has a latent heat of 983 B.T.U. Hence there will be condensed $\frac{328}{983} = .33$ pound.

This leaves $.88 - .33 = .55$ pound of vapor to go to the third effect coil.

The 2.2 pounds of feed comes into the third effect at 192 degrees. The vapor in the third effect is at 126 degrees so that the feed will give up $2.2 \times 66 =$	145.20 B.T.U.
The coil drain from the second effect is at 220 degrees and the vapor in third effect coil is at 192 degrees, so that the coil drain from second effect will give up $.92 \times 28 =$	25.76 B.T.U.
The .55 pound of vapor at 192 degrees has a latent heat of $.55 \times 983 =$	540.65 B.T.U.
The total heat available for evaporation in third effect is then.....	711.61 B.T.U.

The latent heat at 126 degrees is 1021 B.T.U. per pound.

Hence the water evaporated is $\frac{711.61}{1021} = .70$ pound.

This leaves 1.5 pounds to go to the brine overboard pump and a total water evaporated of $.92 + .88 + .70 = 2.5$ pounds water.

We thus have a concentration overboard of about 2½/32ds and an evaporation of 2.5 pounds of water per pound of steam.

EXAMPLE 2. NO FEED HEATING.

As we know so much water will not be evaporated and we want to get about the same concentration of brine overboard, assume 3.25 pounds of feed water per pound steam.

The same temperatures will be found in the various effects as before.

The B.T.U. for evaporation in first effect equals the latent heat of steam at 240 degrees or 952 B.T.U. To heat 3.25 pounds water from 90 degrees to 220 degrees = $3.25 \times 130 =$ 422.5 B.T.U. This leaves $952 - 422.5 = 529.5$ B.T.U. for evaporation. The steam evaporated in first effect shell is then $\frac{529.5}{965} = .550$ pound.

The feed to second effect $= 3.25 - .550 = 2.700$ pounds. This feed gives up $2.700 \times 28 = 75.60$ B.T.U. The B.T.U. available from first effect vapor = 529.50. The B.T.U. available for evaporation $= 75.6 + 529.5 = 605.10$ B.T.U. Latent heat at 192 degrees = 983 B.T.U. Steam evaporated in second effect shell $= \frac{605.10}{983} = .62$ pound.

Feed to third effect = 2.7 — .62 = ...	2.08 pounds
This feed gives up 2.08 (192-126) = ...	137.28 B.T.U.
The heat available from second effect vapor = ...	605.10 B.T.U.
Heat available for evaporation = ...	742.38 B.T.U.

Latent heat at 126 degrees = 1021 B.T.U. Steam evaporated third effect shell $= \frac{742.38}{1021} = .730$ pound. Brine overboard = 1.35 pounds. Water evaporated = 1.900 pounds.

This gives a concentration overboard of about 2½/32 and an evaporation of 1.9 pounds water per pound of steam. The gain due to feed heating is thus seen to be .60 pound of water per pound of steam, an increase of nearly 32 per cent.

To show how much of this gain is due to the vapor heaters interposed between effects, the following example is worked out using the first effect coil drain for heating the feed as much as possible.

EXAMPLE 3. COIL DRAIN HEATING ONLY.

The same temperatures for steam pressures are assumed. As more water will be made with the same concentration overboard we will assume 3.5 pounds of feed. As before, the latent heat in the one pound of entering steam is 952 B.T.U. Of this 3.5 (220-119) = 353.5 B.T.U. are used to heat the feed from 119 degrees to 220 degrees. This leaves 952-353.5 = 598.5 B.T.U. available for evaporation.

The latent heat of evaporation at 220 degrees is 965 B.T.U. Hence there will be evaporated in the first effect shell $\frac{598.5}{965.0} = .65$ pound water.

The feed to go to the second effect is 3.50 — .62 = 2.88 pounds at a temperature of 220 degrees.

In going to the second effect shell where the temperature is 192 degrees the feed will give up 2.88 × 28 = 80.64 B.T.U.

The .62 pound of vapor at a temperature of 220 degrees will give up its latent heat of .62 × 965 = 598.5 B.T.U.

The total heat available for evaporation in the second effect is thus 598.50 + 80.64 = 679.14 B.T.U.

The latent heat at 192 degrees is 983 B.T.U.

Hence the water evaporated in second effect is $\frac{679.14}{983} =$ .69 pound.

The feed to go to the third effect is 2.88 — .69 = 2.19 pounds.

In going to the third effect where the temperature is 126 degrees the feed will give up 2.19 × 66 = 144.54 B.T.U.

The .69 pound of vapor will give up its latent heat of 679.14 B.T.U.

The coil drain from the second effect will give up .62 × (220-192) = 17.36 B.T.U.

The total heat for evaporation in the third effect is thus 144.54 + 679.14 + 17.36 = 841.04.

The latent heat at 126 degrees is 1021 B.T.U. Hence the water evaporated in the third effect will be $\frac{841.04}{1021} = .82$ pound.

The brine overboard will then be 1.37 pounds and water evaporated will be 2.13 pounds (.62 + .69 + .82). The concentration overboard is about 2½/32 and the water evaporated is 2.13 pounds per pound of steam.

The evaporation per pound of steam with the three methods is thus:

	Water per pound steam	Concentration overboard in 32d approximately	Gain due to heaters over no heating	Gain due to vapor heaters over drain heaters
			per cent	*per cent*
No. heating.....	1.90	2.50	...	...
Coil drain heaters	2.13	2.50	12	...
Coil drain and vapor heaters.....	2.50	2.50	32	17

The water evaporated in the various effects with the three systems is as follows:

	1st effect	2d effect	3d effect
No. heating	.55	.62	.73
Coil drain heaters	.62	.69	.82
Coil drain and vapor heaters	.92	.88	.70

The above differences in the distribution of the evaporation shows another gain to be obtained from the vapor heaters aside from the gain in capacity. By reducing the load on the third effect as compared with the first effect the purity of the water is much improved. This is due to reducing the velocity of the vapor from the third effect shell to the distiller and so reducing the tendency of the vapor to pick up water and carry it over unevaporated.

At first glance the use of vapor heaters would seem a case of lifting oneself by his bootstraps. The gain obtained, of course, comes from the added evaporation in the first effect. Steam made in the first effect produces nearly twice as much more when it can be sent along to the second and third effects.

In installing a plant of this type all piping should be put in with as easy bends as possible. Gate valves should be used wherever possible. All joints must be air tight as air leaks cause more trouble than any other factor about the plant. In running pipes from the circulating pumps to the overboard discharge pumps, be sure that these pipes are kept below the circulating pumps so that there will be no vapor pockets in the line.

The following points on operation have been gathered through experience with this type of evaporator and should be looked out for in case of trouble.

Given steam and feed the Lillie evaporator must make water. Little trouble is ever experienced with the steam connections but considerable trouble is sometimes found with the circulation of the feed water. Poor circulation causes falling off in capacity and excessive scaling. It can be discovered by noting pressure gauges where sight glasses are not fitted in the evaporator shells. If the circulation is equally efficient in all shells, the pressure drop through the various effects will be approximately uniform.

If one evaporator develops poor circulation it will be noted that the next higher effect coil and shell pressures become practically the same.

In case of improper circulation over the tubes proceed as follows:

See if the valve in the brine circulating line is open.

Check the speed of the brine circulating pump.

Check the water level.

Examine circulating pump suction line for leaks.

See if the circulating pump glands are properly sealed.

Examine the strainers in the sumps.

Open the priming connection to the brine circulating pump to determine if the pump is vapor or air bound.

The greatest care must be taken to keep evaporators free of air. The necessity for this is even greater than in the case of condensing equipment. In the latter, air leaks mean only a loss of vacuum. In evaporators, besides losing vacuum, the operation of the plant is upset.

To get air out of the steam coils, the plug cocks on the air lines to the back of the steam coils should be opened wide for a few minutes. These plug cocks are fitted with orifices which remain open even when the plug cocks are closed and so allow normal amount of air to be carried off by the air pumps. Open the plug cocks wide on starting and whenever a falling off in capacity seems to indicate excessive air in the steam coils.

In connection with air leaks, the ordinary type of gauge glass fitting should not be used. A type of fitting similar to the Dewrance should be used. This type of fitting allows the packing to be set up without bringing any torsional strain on the glass.

The condensate regulators sometimes stick allowing the steam chests to fill with water. This can be noted from the water gauge glass fitted on the steam chest.

Reduction in capacity will result from cold feed water. The feed water temperatures should be checked frequently to see that the proper rise in temperature is obtained through the various heaters. Keep the distiller overboard discharge temperature as high as possible while still maintaining the proper vacuum in the distillers.

Inspection for scale on tubes and distribution plate should be made frequently until the proper interval for scaling is established. A similar inspection for the sump tanks should be instituted until the interval for cleaning is found.

All salt water piping about the evaporators will gradually form scale. This can be cleaned out by circulating dilute commercial muriatic acid.

High salinity of distillate may be due to one of the following causes: too high a water level in any shell, leakage in distillers (quite rare), vapor or coil drain heater leakage, improper functioning of the vapor separators due to clogging, concentration of the circulating brine too high, excessive splash into the vapor space by improper delivery of the brine upon the distributing plate.

If the brine overboard pump refuses to take suction, prime it through the connection from the distiller overboard discharge line.

Failure to feed properly may be due to the strainers clogging or to insufficient pressure on the distiller overboard discharge line. The latter can be corrected by throttling down on the overboard discharge.

Due to the high vacuum in the last effect shell and the consequent liability of "flashing" in the circulating pump suction chamber the water level in the third effect shell should be carried as high as possible without ebullition in the shell. (Not higher than over the second row of tubes from the bottom.) The added water level helps the suction head on the pump but too high a level will allow the whole shell to fill with a bubbling mixture of vapor and water which will cause priming. It may be necessary at times to reduce the vacuum on the last effect shell to assist the circulating pumps on this effect in maintaining the circulation.

When the evaporating plant consists of two duplicate units both of these should be used continuously. There will be much less accumulation of scale with two units at 5 pounds initial pressure than with one unit at 10 pounds initial pressure. Run twenty-four hours a day on a low pressure rather than a shorter period at a higher pressure. If a sufficient capacity for the ship's needs can be obtained with ten to twelve inches of vacuum on the first effect shell it will be found that the scaling will be eliminated almost entirely.

NOTES ON FOULING OF SHIPS' BOTTOMS AND THE EFFECT ON FUEL CONSUMPTION.

By Capt. Henry Williams (C. C.), U. S. N., Member.

In this paper it is proposed to present some notes on the question of the fouling of ships' bottoms by marine growths and the incidental effect on fuel consumption of cruising vessels. Considering that frictional resistance is the most important element in the resistance to propulsion of practically all ships, it is surprising there has been so little investigation of the possibility of reducing skin friction to a minimum. Ship owners seem to be satisfied that everything possible is accomplished by docking ships, periodically, for cleaning bottoms and painting with anti-fouling paints.

It has been stated that probably more coal is wasted on the bridge than in the engine room. This might be extended to include also foul bottoms of ships. The effort to drive foul ships at full speed has burned many tons of fuel; the normal fuel consumption of all ships is in excess of what their consumption would be with clean, freshly painted bottoms. While, probably, it is not possible to prevent fouling and the consequent increase in fuel consumption, there is room for definite improvement over existing conditions.

In considering this general subject two questions present themselves: (*a*) The reduction to a minimum of frictional resistance, by modifying the conditions of the underwater painted surface; (*b*) The prevention, so far as possible, of fouling growths from forming on the bottoms of ships.

THE SURFACE OFFERING MINIMUM RESISTANCE.

It is supposed generally that skin friction can be reduced by lubrication of underwater surfaces with such materials for example as tallow or graphite. Some manufacturers of ship bottom paints have urged the greasy character of their compositions as an advantage in reducing resistance. The truth of the matter is undoubtedly that such is not the case and that the surface offering the least resistance is one with a hard smooth coating, such as freshly applied varnish or shellac. In a paper read before the Society of Naval Architects and Marine Engineers, in 1915, Naval Constructor McEntee described a Model Basin investigation of the benefits to be expected of polishing or lubricating the wetted surface. The experiments were conducted with friction planes 20 feet long by two feet wide, coated with various lubricating substances and towed at the usual speeds. The result of the experiments indicated no advantage from the use of the materials over a smooth varnished or shellaced surface. It seems evident, therefore, that the most favorable coating for underwater bodies of ships, from the point of view of reducing skin friction to the minimum, is a varnish paint which dries with a smooth hard surface and which will remain in this condition when submerged. There are two causes which, however, may operate to modify this smooth hard surface so as to increase the skin friction when submerged, one, a change in the paint film itself and, the other, the attachment of marine growths.

POSSIBLE CHANGES IN CONDITION OF PAINTED SURFACE.

The fact has been established, with reasonable certainty, that, in some cases, at least, the resistance to propulsion of a vessel begins to increase shortly after the vessel is undocked and before any visible marine growth has formed on the bottom. The amount of such increase in resistance and the rate of increase are not known at present. There are two possible

causes which may account for it, one being a possible change in the paint film, due to being submerged in sea water, and the other being a possible formation of slime or impalpable growth on the bottom. In the discussion of Naval Constructor McEntee's paper above referred to, Sir Archibald Denny stated that at their shipyard, which is on the river Leven, a tributary to the Clyde, they have found an increase in resistance at the rate of nearly 1/2 per cent per day for periods as long as three months. Examination of the bottoms of the vessels in dock revealed no apparent fouling, the surface seeming smooth and even slippery. Shipbuilders in this country have had the same experience in connection with their builders and acceptance trials of new ships, where performance guarantees covered fuel consumption or horsepower for certain speeds. They frequently have found that even a few days out of dry dock made an appreciable difference.

In order to determine the possible change in resistance, after submergence, of the regular Norfolk Ship Bottom Paint, as used on Naval vessels, two steel friction plates were painted at the Model Basin, at the Washington Navy Yard, and kept submerged, one in the fresh water of the tank and the other in a solution of common salt, density 1.026. These plates were towed periodically over a period of about eight months. While the results obtained were not precise, they showed no change in resistance for several weeks and thereafter that the resistance increased until it became about 20 per cent in excess of the first value and then again became constant. There appeared to be no difference in the action of the fresh and the salt water, as far as resistance was concerned.

The plate submerged in fresh water became coated with a thin coat of slime after a few days, but the other plate showed no signs of either animal or vegetable growth. Both plates gradually became roughened by corrosion and it was questioned as to how much of the increase in resistance was due to roughness. The conclusion reached was that the greater

part of the increase in resistance could be ascribed to a change occurring in the paint film itself, aside from corrosion of the metal beneath.

While the above discussion can lead to no conclusive statement of the cause nor of the amount of increase in resistance to propulsion of ships, brought about by a change in the condition of the painted surface, before the attachment of the usual visible growths, it seems evident that under certain conditions such an increase actually does occur. It is desirable to determine the facts with a view to finding a remedy. Studies are being made both by towing friction plates in the Model Basin and by microscopic studies of the films of various paint compositions after different periods of submergence in sea water. It obviously is desirable in designing new ship bottom paints, that consideration should be given to the above, the most desirable paint being one which, presenting a smooth, hard surface at the outset, has the qualities, among others, of maintaining its film unchanged after submergence in sea water for the usual periods.

FOULING OF SHIPS' BOTTOMS BY MARINE GROWTHS.

In Model Basin experiments described in Naval Constructor McEntee's paper referred to in the foregoing, tests were made to determine the increases in frictional resistance, due to the presence of actual fouling growths. Friction plates were placed in Chesapeake Bay near Norfolk and were removed periodically, packed so as not to disturb the fouling and sent to the Model Basin for frictional resistance tests. It was shown that at a speed of 10 knots there was a maximum increase in resistance such that the resistance of the plate when foul, was four times as great as when it was clean and freshly painted.

The relative effect of fouling on the actual frictional resistance of a ship, probably, would not be so great. There is

ample evidence of a considerable increase in resistance and resulting decrease in speed of a ship, but there appear to be no precise data as to the rate or amount of such increase in resistance to propulsion of ships nor of the amount of the resulting increase in fuel consumption. The Department recently has assigned a destroyer to make periodic standardization runs over the measured mile course at Rockland for the purpose of determining accurately the amount and rate of increase in power necessary to maintain the different speeds as the time out of dry dock increases. The battleship *Tennessee* is being standardized periodically on the West Coast for the same purpose. While it is expected that the data obtained will be of value in showing the amount of increase in power consumption of the ships in question under the special conditions existing in each case, the results will be by no means conclusive. Before definite conclusions can be drawn, it may prove necessary to standardize ships operating or lying in different waters at different seasons of the year and possibly with different anti-fouling paints on their bottoms. Data which would show for any ship, under various different conditions, the increase in power consumption, resulting from fouling of the bottom, would be of value in considering tactical questions and might prove of the utmost importance in a campaign. It is a question of which there ought to be full information available and it should be studied until thoroughly understood in all its various ramifications, including the most important of all, which is, how the attachment of marine growths to ships' bottoms may be prevented.

THE PROCESS OF FOULING OF SHIPS' BOTTOMS.

The marine organisms which attach themselves to the bottoms of ships and thus comprise what is known as fouling, are principally barnacles, hydroids, algae and bryzoa. Oysters and mussels are sometimes seen, but not usually on cruising vessels.

The following is a definition of the barnacle given by Huxley: "The barnacle may be said to be a crustacean fixed by its head and kicking the food into its mouth with its legs." The name barnacle comes from one of the most extraordinary and persistent myths of the middle ages which attributed to barnacles the origin of the "bernacle goose." This bird is a winter visitor to Great Britain and its Arctic nesting place being then unknown the idea became current that it originated within the shells of the so-called goose-neck barnacles which grow along the sea shore.

Barnacles which belong to the crab family have a rather complicated life history. The young, called nauplii, barely visible to the naked eye in strong light, are free swimming and resemble minute water fleas. The nauplius, which moults several times, at the last moult is transformed into a bivalve, like a minute mussel, known as a crypid. Protruding from the shells of the cryprid are antennules provided with sucker like discs, to which are connected cement glands, serving for attachment. There are other projections which include a large olfactory filament. The cyprid attaches itself at first temporarily by means of the disc and other filaments and then permanently by means of cement. It then undergoes its final moult and becomes a barnacle with rudimentary shells. It attains a diameter of one-eighth of an inch in about two days and grows rapidly from that time on. The adult barnacle is attached by cement to the surface and with each period of growth new cement glands are developed, the entire surface of the base in most species being cemented in place. The barnacle subsists on minute organisms in the sea water, drawn in by feather like legs, which it extends and waves back and forth to create currents of water. The name cirripedia, which is used to describe the broad types, means "curl footed" and refers to the legs, which are used only in feeding. The shells of some barnacles consist of separate plates which are moved back and forth when

it is feeding and also when ejecting its young. In other forms the shells are in one piece, except two plates which move back and forth.

Barnacles are usually bi-sexual; each barnacle fertilizes its own eggs, which it retains within the shell cavity until hatched and, frequently, until after the first moult when the young issue forth in clouds and begin the free swimming stage, undergoing the metamorphosis described above. There are a large number of different species of barnacles, the one most commonly found on ships' bottoms being the so-called acorn barnacle, *balanus eberneus.*

Hydroids are sometimes called "zoophytes" because they resemble plants in general appearance, but are in reality animal organisms. It is the growth of hydroids on the bottoms of ships that frequently is referred to as "grass." The eggs of the hydroids after fertilization, develop into larvae, known as planulae, and in that form become attached to any surface, such as a ship's bottom, and produce first a single stalk or stolon and then a number of stems. On the end of the stems polyps are formed which develop into small free swimming animals, like the so-called jelly fish. There are thus two generations, only one of which is seen attached to the bottom of ships.

Algae, which are in reality sea plants, generally are not found on ships' bottoms except along the water line. They are produced from spores, which are for a time free swimming and later find attachment to a free surface and send out their growth. The spores are provided with a number of long hairs which whip back and forth, resulting in movement, and as these spores tend to move toward the light, growths of algae most frequently are seen near the waterlines of ships.

Bryozoa frequently form a portion of the fouling matter found on ships' bottoms. The term means moss-animals and is applied because of the resemblance to a patch of moss. These animals are found in colonies, spreading out radially

from a center, and are usually only a single layer in thickness. They live and feed in much the same way as barnacles, but are much smaller. The young are discharged as free-swimming larvae which have eye spots and are sensitive to light. After a free-swimming existence of a day or two, they attach to some surface and transform into the adult form. By means of a rapid process of budding, a single individual builds up a large colony in a short time.

Certain worms also are found on ships' bottoms. These worms live in white coral-like tubes, which are hard and are often as much as two or three inches long. The young of these animals also are free living for some time and are sensitive to light. The larvae go through a complicated series of moults before reaching the condition of the adult.

It is known that in northern waters, the fouling growths reproduce only during the spring and summer months. While there is not complete information, the fact would appear to be of importance in considering the fouling of ships. For example, a ship, if freshly painted with anti-fouling paint, that would remain effective during the season of reproduction, need not be redocked until the following spring, if it remains in northern waters. If redocked sooner, the poison in the paint would lose its effectiveness before the growths began to reproduce and thus the ship would be more liable to fouling.

PREVENTING THE FOULING OF SHIPS' BOTTOMS.

A clear glass plate submerged in barnacle infested waters, within the space of a few hours, on examination, will show attached to it a number of cyprids and perhaps other growths. The same or similar plates coated with paints containing any of many different substances, under the same conditions, will go for periods up to six months or more without showing the attachment of cyprids or other growths. The length of time that fouling can be prevented, in infested waters, depends ap-

parently on the materials used in the paint. The problem which has presented itself to shipowners, since the earliest days, is to determine what materials should be applied to the bottoms of ships to prevent the attachment of marine growths. For wooden craft the problem is solved reasonably satisfactorily by the use of copper sheathing, which, though intended primarily to prevent the destruction of the wood bottoms by borers, serves to keep the bottoms fairly clean. As the use of sheathing is not practicable on steel ships, these must depend upon paint, necessary also to protect the steel from the corrosive action of sea water.

For many years the problem of determining the best substances to include in ship bottom paints to prevent the attachment of fouling growths, apparently, has been attacked in a purely empirical manner. No scientific studies appear to have been made, of the manner of attachment of marine growths, nor of the means by which they might be killed shortly after attachment. All investigations, of which any record can be found, consisted of experimentation with different paint compounds, with a view to the use of whichever seemed most effective after experimentation.

In order that the question may be understood thoroughly and all means of preventing fouling developed fully and to permit a thorough understanding of the technological questions involved in the production of a durable bottom paint containing poisonous or other ingredients to prevent fouling, it is necessary to undertake also a biological study of the fouling organisms. We should understand their method of operation, their means and manner of attachment and their reaction to the substances which we already know prevent their attachment and to other substances which might be useful. Microscopic studies must be made of the larvae of the various organisms and their reactions to various conditions and substances. Studies must be made of known phenomena in order that there may be a thorough understanding of their causes.

Fortunately it has been possible to start an investigation along the above lines. The Navy Department has been able to secure the cooperation of the Bureau of Fisheries of the Department of Commerce and the use of their laboratory at Beaufort, N. C. It has been able to secure the assistance of a leading paint technologist and a leading biologist to conduct investigations. It has secured also the cooperation of the Chemical Warfare Service of the Army in the investigation of the question of determining the most effective toxic material to include in the paints. This work has been going on for about one year and progress has been made in gaining an understanding of the biological questions and in eliminating many proposed toxic materials found to be ineffective and unsuitable. A large number of compositions have been tested by applying them to steel plates for submergence and progress has been made in solving the technological problems involved in the designing of ship bottom paints. While no conclusive answers to the questions recited in the foregoing can be given, it may be of interest to state here the present practice in regard to prevention of fouling and to speculate upon the causes of the various known phenomena.

As noted heretofore, when copper sheathing for wooden ships was adopted, though primarily for preventing the destruction of the ships' bottoms by marine borers, it served also practically to prevent fouling. When, however, iron and steel ships came into use the problem of preventing fouling as well as that of preventing corrosion became very pressing. Contemporaneous literature described many cases of excessive fouling or corrosion, or both, and ships on distant voyages were handicapped greatly and delayed by the mass of marine growth accumulated, which science at that time offered no means of preventing, except that of sheathing with copper. This was tried and continued to be tried up to recent years, with invariably resulting corrosion, from electrolysis, of the steel hull. Sheathing with zinc plates was tried, but while it

did not cause corrosion, it did not prevent fouling. In the early days of iron shipbuilding the "stand-patters," favoring wooden ships, urged against their use the susceptibility of iron ships to fouling.

It soon became evident that the solution to the difficulty lay in painting the bottoms of the iron or steel ship with a composition that would have the property not only of protecting the metal of the hull from the corrosive action of the seawater, but of preventing the attachment of the marine growths. At that time began the search, which continues to the present day, for an effective anti-fouling paint. The question of preventing corrosion has been solved fairly satisfactorily and, except for vessels with light bottom plating, serious corrosion of underwater plating of ships, generally speaking, does not take place. The paints used, protect satisfactorily from corrosion longer than it is possible for ships in operation, to go without drydocking. Ships lying-up must be docked at such periods as necessary to prevent deterioration by corrosion, and in such cases there is no object in preventing fouling. Marine growths appear to assist in preventing steel from corroding.

It is now the practice almost universally in painting the bottoms of steel ships to use two kinds of paint, the first coat being called anti-corrosive, and intended as a priming coat for the anti-fouling paint which is applied over it. As ships must not be delayed in dry dock to await the drying of the bottom paints use is made generally of quick drying spirit-varnish paints. Suitable pigments are added to give the necessary body and to the anti-fouling paint is added the toxic substance intended to prevent the attachment of the various growths. There are some exceptions to the use of spirit varnish paints, as there are well known anti-fouling paints which, when cold, are thick and heavy and must be heated for application. Even with these, which are called grease or soap paints, spirit-varnish anti-corrosive paints are used.

The toxic substances almost invariably used are salts of mercury, copper, arsenic or zinc. The salts of mercury, for a long time found to be the most effective, unfortunately are expensive and in wartime become almost unobtainable, due to the demand for mercury for other purposes. Copper salts are used extensively for the reason that they are cheaper and, in some cases at least, have been shown to be fairly effective. It is probable that copper and mercury salts in ship bottom paints cause some corrosion of the steel hulls, by electrolytic action. However, this never has been shown to have had serious consequences and little account usually is taken of this possibility.

There is no conclusive information as to the manner in which the various toxic or other materials in bottom paints act to prevent the attachment of the growths. Whether the larvae are killed after attachment or whether they are prevented or discouraged from attachment, is not known. It is of interest, however, to speculate on the possibilities. In the case of copper sheathing the so-called exfoliation theory has been accepted. This is that the growths become actually attached, and the copper sheathing being in continual state of wearing away, or exfoliating, throws them off with the wasted copper. Present knowledge, however, indicates that the process is essentially different, being more one of directly poisoning or discouraging the young growths either before or after attachment. Sea water is a solvent for all substances and contains many chemical compounds. Its action on metals consists of converting them into more or less soluble salts, rendering them poisonous to the growths. As has been noted in the foregoing, cyprids have an olfactory filament, which probably enables them to determine in advance the fitness of their food and the suitability of any surface for a resting place. It seems probable that this sense of smell or "anticipatory taste" causes them to keep away from surfaces coated with poisonous or unpleasant materials which are throwing off a solution. It

is reasonable to expect that besides the known metallic poisons mentioned in the foregoing, it will be possible to find other substances that have an equal or greater repellent effect on the young organisms. Comprehensive biological studies are being conducted to determine the effect of the various poisonous substances on different species of organisms, the manner in which the effect is caused and the most advantageous method of applying the substances determined to be most effective. However, the question is exceedingly complicated and involved and will not be solved without a great amount of investigation and work.

There are found frequently upon the bottoms of vessels in dry dock numbers of dead barnacles and other dead organisms. An interesting example of this occurred recently in the case of the *Maryland,* which was docked at the New York Navy Yard shortly after return from a short stay at Rio de Janeiro. The bottom appeared to be clean and free from growth but upon close examination there were found to be attached to the bottom a number, about 50 to the square foot, of small dead barnacles less than 1/8 of an inch in diameter. These presumably had become attached while the vessel was in Rio harbor, but whether their death was caused by the poison in the bottom paint, by change in temperature and character of sea water or by the pollution in the harbor water at New York is not known and only can be conjectured. In the case of the *Nevada,* which spent some months in the harbor of Rio, the vessel was reported to be very foul. Upon docking at Norfolk it was found that all growths on the bottom were dead and many had dropped off. Only the green algae, which was extensive along the water line, was living. The biologist who inspected the growths on the *Nevada* reported that the common acorn barnacles were not present, but that there were numerous barnacles of two types not previously seen by him on bottom of ships.

Another phase of this question lies in the possibility of killing and removing fouling growths by taking the vessel into fresh water. Naval ships when passing through the Panama Canal have been held at times in Gatun Lake for this purpose. Unfortunately there is no definite information as to the length of time in fresh water necessary to kill the growths.

One of the most interesting developments in the experimental work at Beaufort, not only as to its possible utility, but also as showing the lack of precise information on conditions governing fouling, relates to the effect of colors in the fouling growths. It was suggested, as a result of experience in painting live boxes, by the biologist in charge of the aquarium at Bermuda, that certain colors are repellent to marine growths. Accordingly panels were exposed at Beaufort painted different colors, white, yellow, green, red, blue and black. The paints contained no poison and were identical, except as to the pigments used to obtain the different colors. After exposure for three months inspection showed the white panel practically free from fouling, the yellow and green slightly foul and the red, blue and black heavily coated with fouling growths. An interesting feature is that the red iron oxide colored paint, which is almost the conventional color for anti-fouling paints, showed a decided tendency to attract fouling growths. These results can not be considered conclusive and steps have been taken to paint, experimentally, the bottoms of several ships, in half sections, with the regular Norfolk anti-fouling paint and with exactly similar paint, except that the red iron oxide pigment is replaced with a white pigment. The results of these tests together with the additional experiments at Beaufort should show whether fouling can be reduced by the use of light colored paints.

SHIP BOTTOM PAINT USED ON NAVAL VESSELS.

Prior to 1909, the bottoms of naval vessels were painted with different approved brands of bottom paints, purchased

from their respective manufacturers on proprietary requisitions. Under the rules existing at that time change was not made in the brand of paint used on the bottom of a particular ship, from that previously used, except by special order of the Department. Difficulties arose continually through the efforts of manufacturers of paints, not approved by the Department, to secure such approval. The cost of paints as purchased without competitive bidding was higher than seemed justified and the results were not always satisfactory. For these and other reasons the Department decided to undertake experiments with a view to determining a formula from which to manufacture its supplies of bottom paints in Navy Yards.

The Norfolk Yard developed formulas, which, after being tested fully in comparison with the commercial brands of paints then in use, were approved in 1909 to be used in the manufacture of bottom paints for naval vessels. The so-called Norfolk ship bottom paints are used when available on all naval vessels and are manufactured at the Navy Yards Norfolk, Mare Island and Cavite.

The experience of the Shipping Board in the purchase of ship bottom paints for its vessels was similar to that of the Navy Department. The purchase of proprietary paints led to much difficulty and unsatisfactory results, and sometime ago it was decided to adopt the Norfolk ship bottom paints for their vessels and their requisitions now specify the composition of the paint to be in accordance with the Navy standard formulas. The results obtained are reported to be highly satisfactory to the officials of the Shipping Board charged with the maintenance of their ships.

Naval vessels docking at foreign stations frequently are forced to use the nondescript patent ship bottom paints offered by these dock yards. The results, generally speaking, are unsatisfactory and the cost greater than that of the standard naval paint.

THE INTERVAL BETWEEN PERIODIC DRY DOCKING OF VESSELS.

The Navy Regulations require that Naval vessels in commission shall be docked at intervals of not less than one year, but may be docked, if convenient opportunity offers, at intervals of about six months. Any extension of docking interval beyond one year, for a ship in commission, must be authorized by the Secretary of the Navy. In the interest of fuel economy the tendency is to decrease the docking interval, while the need for economizing on docking costs calls for extending the docking interval. There is, of course, in each case a point where it becomes uneconomical not to dock and clean a cruising ship, due to the increase in fuel consumption caused by the foul bottom.

On the other hand, there is a tendency to exaggerate the magnitude of possible saving in fuel and to insist on unnecessarily short docking intervals. It is a fact that no one can tell in advance of docking, in the present state of knowledge, what is the condition as to fouling of a ship's bottom, except, of course, in extreme cases of very heavy fouling which becomes obvious. A ship may cruise for a year and at the end of that time have a comparatively clean bottom, while another ship may show heavy fouling at the end of a few months. There is, of course, a law governing the growths of fouling on ships' bottoms, the variable factors being the kind of antifouling paint used, the season of the year, the amount of cruising performed and the waters visited by the vessel. An effort is being made at the present time to deduce laws governing this question, through the inspection while in dry dock of a large number of vessels. This inspection is being made by a biologist familiar with marine growths and it is hoped that his observations may lead to conclusions useful in determining laws governing the growth of fouling. It has been suggested, in connection with this study, as a means to supply information as to the growths in the various harbors visited, that

certain ships be supplied with test panels for exposure in the waters of the harbors in which they anchor. These panels, after exposure for a certain length of time, would be sent to a biological laboratory for examination of the attached growths. This might permit determining the source of the growth later found on the ship at docking and assist in reaching conclusions as to the conditions permitting or preventing this attachment and growth.

The most important phase of the question of a proper docking interval is the effect on military operations of the Fleet. In case of a campaign away from home bases, as for example in the Pacific, due to limited docking facilities, probably it would not be possible to dock ships except at long intervals or when necessary for repairs. Foul bottoms might prove a serious tactical handicap. It seems important, therefore, that in order to develop in peace time the full possibilities, ships should operate with extended docking intervals. The determination of the most effective anti-fouling paint is a question of first military importance.

In the effort to determine the progressive increase in consumption of fuel caused by the change in condition of the wetted surface and by the accumulation of fouling, a large amount of data has been collected and presented by various cruising ships. Generally speaking these data are based on gross fuel consumption on different runs and at various periods of varying length between dockings. While the data thus gathered show clearly that substantial and progressive increase in fuel consumption takes place as the time out of dry dock increases, so far presented they are not conclusive. Besides fouling, the elements causing variation in fuel consumption are important and not susceptible of determination. They consist chiefly of variations in quality of fuel, condition of machinery and of operation, weather conditions and temperature of circulating water.

An important consideration affecting the docking interval is the effect on Engineering competition runs of a foul bottom. Ships freshly docked prior to competition runs are at a decided advantage in fuel consumption over ships longer out of dock. This results in continued pressure on the part of these ships for special docking in anticipation of such runs. It is not always possible for ships to be so docked and it would seem desirable to devise some method by which allowances could be made depending on the probable increased fuel consumption due to this cause.

THE LANGLEY CHRONOGRAPH.

By Lieut.-Commander Guysbert B. Vroom, U. S. Navy,
Member.

This instrument was designed and assembled by the Engineer Department of the U. S. S. *Langley;* parts necessary to adapt the standard apparatus were manufactured in the ship's machine shops.

The purpose of the instrument is to provide means whereby the action of aeroplane launching catapults, as installed on the *Langley,* may be analyzed. The specific data supplied by the chronograph are two curves drawn simultaneously on an indicator card; (*a*) a curve of cross-head displacement, and (*b*) a curve of impulse cylinder pressures; the X-axis gives time-increments, since the indicator recording-cylinder is driven continuously at a constant speed, which may be accurately regulated within very small limits.

The curve of impulse cylinder pressures is obtained by the ordinary indicator mechanism; the cross-head displacement curve is obtained through specially designed mechanism, which reduces the travel of the cross-head to the travel of the recording pencil, superadded to the indicator gear.

DESCRIPTION.

The apparatus consists of a spring motor (B, Fig. 2) on the structure of which is mounted a standard, outside spring, Thompson Improved Indicator (C, Fig. 2). The Indicator is so mounted that a gear wheel (D), which is secured to the

recording cylinder of the indicator, meshes with a gear wheel of the gear train of the spring motor. The recording cylinder (A) turns continuously, as noted above; the spring against which it works when used for indicating reciprocating machinery, having been removed.

The speed of the recording cylinder is controlled, and may be varied, by the governor (*a*, Fig. 2) on the gear train, which is regulated by a screw (*b*, Fig. 2).

The pressure cylinder of the indicator is connected to the impulse cylinder of the catapult by piping, and the pencil mechanism of the indicator therefore records the impulse cylinder pressure throughout the stroke, and the pressure at any instant can be read off from the card.

To obtain the cross-head displacement curve a reducing mechanism with a recording pencil is used. Referring to Figs. 1 and 2, E is a drum, mounted on the foundation of the apparatus, on which 24 grooves are cut. The diameter across the bottom of the grooves is 3 inches. Through gears, this drum drives a small spindle (F, Fig. 1) upon which is cut a spiral groove such that the diameter of the spindle across the bottom of the groove is 0.17 inch.

The ratio of reduction, drum to spindle, is 4 to 1.

Mounted on the indicator frame is a slotted arm, (*m*, Fig. 1), upon which rides a pencil holder (*g*, Fig. 1). A cord secured to *g* passes over a fair-lead *f* and is secured to the spindle F. The large drum E is driven by a cord secured to the catapult cross-head; therefore the motion of the catapult cross-head is transmitted to the recording pencil *h* (Fig. 1), such that the final ratio of cross-head displacement to pencil dispacement is 196.75″ to 2.75″. Drum E works against a spring (G, Fig. 2) which stops the drum instantly at the end of the stroke, and takes up the slack of the cord which actuates the drum on the return stroke of the cross-head. (Note: Pencil *h* also works against a spring.)

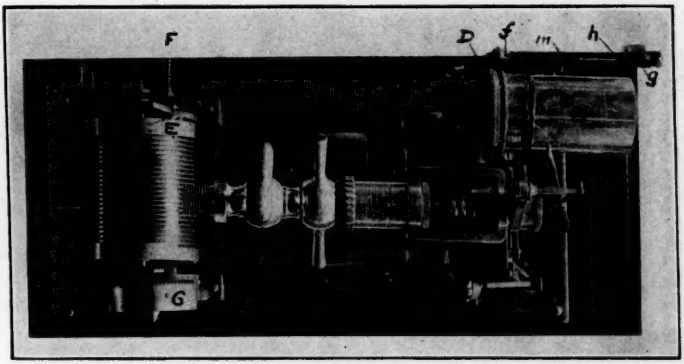


FIG 1.

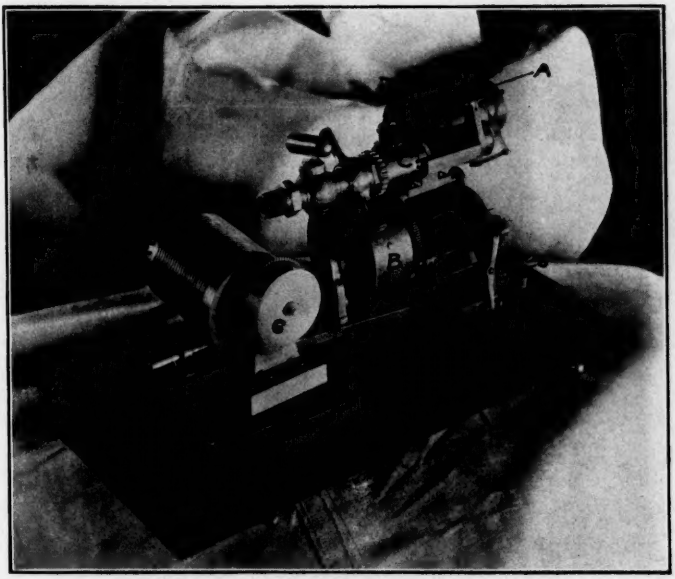


FIG. 2.

USES.

1. For checking the acceleration of cross-head, as actually obtained with empirical valve settings and air-pressures, against the ideal acceleration curve;

2. For calibrating the catapult, and constructing a table of starting pressures and brake pressures, to be used for obtaining a given final velocity with all types of planes.

To illustrate the data obtained from the acceleration curve, the curve as drawn by the chronograph is shown in Fig. 3. Since the cylinder revolves at a known rate previously determined, the reference line O-X, drawn by the pencil *h* when the chronograph is started, represents the time in seconds for one revolution of the recording cylinder of the indicator. When the catapult cross-head makes its stroke, the displacement curve is drawn; point *n* is the end of the stroke, and the distance X*n* is the total displacement of the cross-head, to the scale determined by the design of the reducing mechanism. *mn* = time for complete stroke.

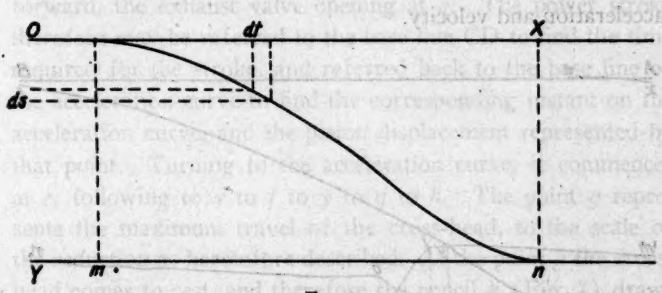


FIG. 3.

To find the acceleration at any instant, take the increments *ds* and *dt* on the curve, choosing any desired value of *dt* to the scale of the total length of O-X, and, by dropping perpendiculars to the curve from the points on O-X and projecting the intersection of the perpendiculars with the curve to O-Y, read off to scale the value of *ds*. Then, having *ds* and *dt*

$$\frac{ds}{dt} = dv$$

$$a = \frac{dv}{dt}$$

The theoretical curve, plotted from the equation

$$S = \frac{Kt^2}{2} \; (K = 32 \text{ [approx.]})$$

will give a comparison of actual against theoretical performance. Impulse cylinder pressure curve is not shown on this Figure.

Figure 4 shows a complete card obtained on a trial of the catapult. Its analysis gives all necessary data regarding the functioning of the valves, such as the time required for valve opening; time required to take up the slack in the cable before the car begins to move; the length of the power stroke; power required to obtain a given average acceleration and final velocity, from which the pressure required for the launching of other weights may be predicted; average acceleration; instantaneous acceleration and velocity.

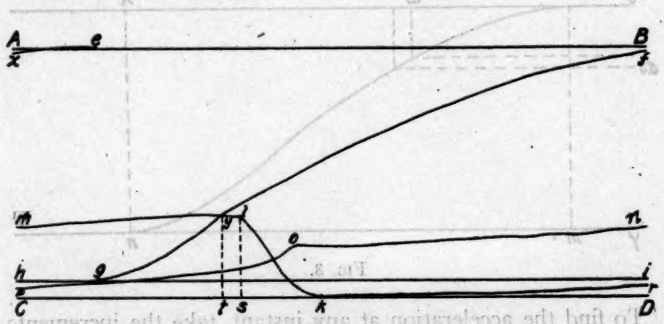


FIG. 4.

Referring to Fig. 4, the speed of the recording cylinder was regulated to 3 seconds per revolution. Therefore the line AB represents 3 seconds and, since it is 6.3 inches long, 0.1 inches represents 1/21 seconds.

The instrument is started before the catapult is fired, and the two lines AB and CD are drawn by the pencil *h* (Fig. 1) and the indicator pencil, respectively. These lines are the base lines, or lines of reference, for the acceleration and pressure curves respectively. The rotation of the recording cylinder is continuous, and until the catapult is fired the pencils simply move over the lines AB and CD.

At the instant of firing, however, the acceleration and pressure curves are drawn, starting from points *e* and *k* (Fig. 4). The curves will be considered separately.

The pressure curve is *klmnoprk*. With reference to the base line, which represents atmospheric pressure, the instantaneous pressure can be measured by scale, according to the scale of indicator spring used. Point *l* shows the maximum pressure; from *l* to *y* the pressure remained constant showing that the length of the line *ly* (to the scale 1/10 inch is equal to 1/21 second) is the time required to take up the slack in the cable. From *y* to *m* to *n* to *o* the pressure falls as the piston moves forward, the exhaust valve opening at *o*. The power stroke therefore may be referred to the base line CD to find the time required for the stroke, and referred back to the base line of the acceleration curve to find the corresponding instant on the acceleration curve, and the piston displacement represented by that point. Turning to the acceleration curve, it commences at *e*, following to *x* to *f* to *y* to *g* to *h*. The point *g* represents the maximum travel of the cross-head, to the scale of the reduction as heretofore described. At the point *g* the cross-head comes to rest, and therefore the pencil *h* (Fig. 1) draws a straight line *hgi* parallel to AB. The complete acceleration curve is therefore *exfyg*. To the scale of 1/10 inch equals 1/21 second, the time required for the stroke is *e*A plus *ig*. With *e* and *k* as the starting points of the acceleration and pressures curves respectively, and using the above time-displacement scale, corresponding points on the two curves may be found. (It must be remembered that *e* corresponds to *y*

in point of time, since the pressure is building up from *k* to *l*, and the line *ly* represents the time required to take the slack out of the cable before the car begins to move (point *e*).

To find the instantaneous velocity at any point on the acceleration curve it is only necessary to drop a perpendicular from AB to the curve and measure the perpendicular. This distance in inches, multiplied by a constant, gives the displacement of the car (the constant is calculated from the reduction ratio of the instrument); the time element is measured on AB from the origin *e* to A, and from B to the selected point, to the scale 1/10 inch equals 1/21 second.

Then $V = \frac{2s}{t}$

The average acceleration is found from $s = 1/2at^2$.

The points on the acceleration curve representing maximum velocity may be found by drawing the tangent to the curve from *hi* as a base line. These points may be referred to the pressure curve by dropping the perpendicular to the base line AB and, having determined the elapsed time from the origin, using the elapsed time to the same scale on the pressure curve base line CD, dropping the perpendicular to the pressure curve, to find the pressure corresponding to a maximum velocity.

In Fig. 4, if a perpendicular is dropped from *l* to CD, *ks* = time required for valve to open and pressure to build up; likewise *st* = time required to take up slack of cable; in further analyzing the curves, it can be shown that point *o* on the pressure curve, at which the pressure is released, corresponds to the point on the acceleration curve showing maximum velocity.

It should be noted, in examining Fig. 4, that the card differs from the ordinary indicator card in that the rotation of the recording cylinder of the mechanism is continuous, and not reciprocating; and that the curves are not closed curves. That part of the paper containing the space for notes is overlapped when the paper is on the cylinder, and therefore points *x* and *f*, *m* and *n*, *p* and *r*, *h* and *i* are identical.

NOTES.

FEED HEATING FOR HIGH THERMAL EFFICIENCY.

ECONOMIES OF 25,000-KW. POWER STATIONS USING SINGLE- AND MULTIPLE-STAGE CONDENSER HEATERS, WITH AND WITHOUT ECONOMIZERS, DETERMINED FOR THE PURPOSE OF DEMONSTRATING THE EFFECT OF VARYING THE FEEDWATER TEMPERATURE.

By Linn Helander,* East Pittsburgh, Pa.

For power plants using single- or multiple-stage feedwater heaters of the condenser type, the temperature of the boiler feedwater as it leaves the heaters should not be less than 150 degrees F. when using economizers and probably not more than 260 degrees F. when not using economizers, although certain conditions permit improving thermal efficiencies up to a temperature of 300 degrees F., corresponding to a pressure of 72 pounds per square inch absolute in the heater. The maximum of this range was established by consideration of fuel charges only. The lower limit of 150 degrees F. was chosen to avoid mechanical difficulties met when sending colder water to economizers, although this temperature, as will be seen, is less than the lowest temperature justified by purely thermal analyses of the 25,000-kw. plant used as a basis for the present studies.

That there exists for any fixed set of conditions a definite feedwater temperature at which the efficiency of power generation is a maximum is most readily demonstrated by consideration of a theoretically perfect generating plant using perfect condenser heaters. Since the feedwater heaters are considered as being perfect condensers, the pressure within them, and for the purposes of the theoretical analyses, the back pressure on the auxiliary turbine or at the extraction point on the main unit supplying the heating steam will be that corresponding to the vapor tension of the feedwater, and so will increase as the temperature of the feedwater increases. This increase in pressure in turn will increase the water rates of the steam units exhausting to the heaters, or, from another viewpoint, will decrease the work capable of being done by a pound of steam used for heating the feedwater.

Consequent upon increasing feedwater temperatures, therefore we have a reduction in the amount of work obtainable from each pound of steam used for heating the feedwater and an increase in the total amount of steam required for heating purposes. The relation between these offsetting effects is such that the work done by the steam used for heating the feedwater increases as the temperature is increased to a certain point, after which it decreases. This is also shown by curves of Fig. 1. Any increase in feedwater temperature beyond that for which maximum work obtains continues to augment the demand for exhaust steam, but the capacity of a given quantity of the steam to do work is so reduced that the result is diminution in the total power generated by it.

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Contributed by the Power Division and presented at the Annual Meeting, New York, December 4 to 7, 1922, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

EFFECT OF MULTIPLE-STAGE HEATING.

A conception of what happens may be had by reference to the temperature-entropy diagrams, Figs. 2 and 3, given for both single-and double-stage heating, assuming for simplicity that saturated instead of superheated steam is used.

As the number of stages of feedwater heating increases, the work derived from the steam used for heating the feedwater increases and the temperature of the feedwater, as established for maximum theoretical efficiency,

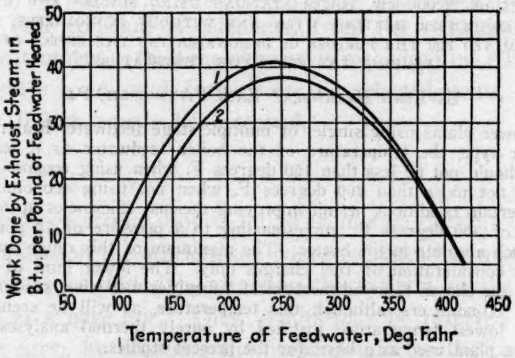

FIG. 1.—WORK DONE BY EXHAUST STEAM.

(Curve 1 shows for single-stage heating that the theoretical maximum work obtained from steam subsequently used for heating boiler feedwater is obtained with a final feedwater temperature of approximately 240 degrees F. when the initial temperature of the feedwater is 79 degrees F. Curve 2 is based on a condensate temperature of 92 degrees and shows that in this case the maximum work is obtained with the final feedwater temperature at 260 degrees. Initial steam pressure, 325 pounds; superheat, 200 degrees F. The peaks of the curves establish the feedwater temperatures for maximum theoretical thermal efficiency. With two-stage heating and the same initial steam condition (condensate at 79 degrees F.) the maximum work is obtained at a feedwater temperature of approximately 300 degrees, which theoretically, therefore, is the temperature for best thermal efficiency.)

approaches that of the initial steam. Using an infinite number of stages, the temperature of the feedwater for best efficiency is equal to that of the initial steam and the efficiency of the theoretical power-generating cycle is that of Carnot's cycle.

DESCRIPTION OF ASSUMED POWER STATION.

Illustrative of what is involved in the practical problem of determining the most efficient feedwater temperatures for power stations, the effect of varying this temperature was determined for several assumed stations of 25,000-kw. capacity, using various methods of heating the feedwater. To illustrate the influence of factors such as the water rate of the main unit and the slope of the Willans line of this unit, the internal Rankine-cycle efficiency of the bled steam and the Rankine-cycle efficiencies of the house turbine, two cases—designated Case 1 and Case 2, respectively—were worked out for each arrangement of feedwater heating. The Rankine-

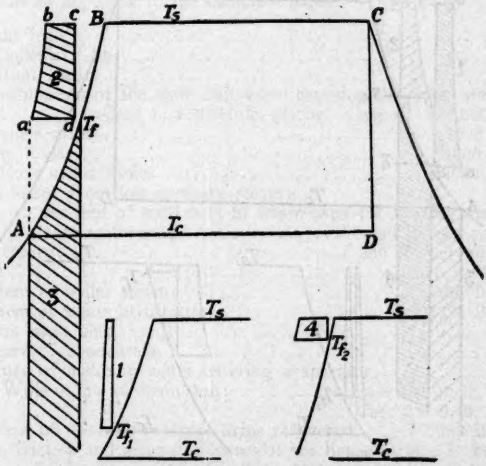

FIG. 2.—TEMPERATURE-ENTROPY DIAGRAM FOR SINGLE-STAGE HEATING USING DRY SATURATED STEAM.

(Large area *ABCD* represents the work done by the steam passing through the main unit. The shaded area 2, above the water line, represents the work done, before entering the heater, by the steam used for heating the feedwater. The shaded area 3, below the water line, represents the heat added to the feedwater and is equivalent to the area under line *ad*. As indicated by areas 1 and 4 of the diagram at the bottom of the illustration, the area representing the work done by the steam used for heating the feedwater becomes rather small when the feedwater temperature deviates largely from that for best efficiency. The letters with subscripts f, f_1 and f_2 indicate the feedwater temperature for different positions of the area showing the work done by the steam used for heating the feedwater. The temperature of the initial steam is Ts; Tc is the temperature of the condensate.)

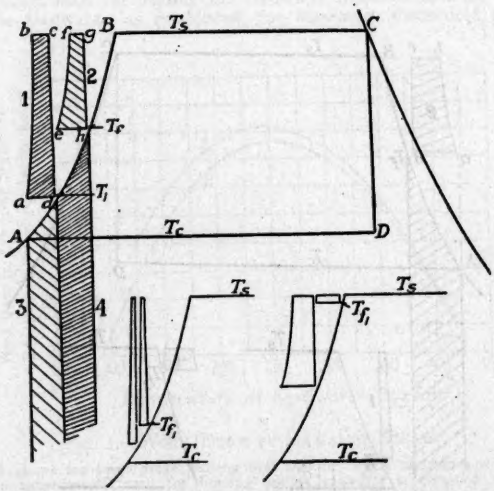

FIG. 3.—TEMPERATURE-ENTROPY. DIAGRAM FOR DOUBLE-STAGE HEATING USING DRY SATURATED STEAM.

(Large area *ABCD* represents the work done by the steam passing through the main unit. Instead of a single area above the water line, as for single-stage heating, two areas, *abcd* and *efgh*, represent the work done by the steam passing respectively to the first and second stages. For equivalent heating effects in each heater, areas 3 and 4, below the water line, are equal. The figures at the bottom of diagram show that when the feedwater temperature is rather close to either the condensate temperature or the temperature of the boiler steam, the sum of the areas representing the work done by the steam used for heating the feedwater is small. When the temperature of the feedwater equals that of the boiler steam the effect is simply that of a single-stage heater. Evidently at some temperature between that of the condensate and that of the boiler steam the sum of the areas representing the work done by the steam used for heating the feedwater is a maximum. The temperatures indicated as Tf, Tf_1 and Tf_2 are those of the feedwater.)

cycle efficiency used in Case 1 for the bled steam based on the steam pressure on the turbine side of the throttle was approximately 67 per cent, slope of the Willans line was 12 pounds per kw-hr., and the water rate of the main unit when carrying the total gross station load was 10.6 pounds per kw-hr. For Case 2 the corresponding values were 80 per cent for the Rankine-cycle efficiency of the bled steam, 9 pounds per kw-hr. for the slope of the Willans line, and 10.26 pounds per kw-hr. for the water rate of the main unit.

The following are the principal data used as bases for the heat-balance study of the paper. The various formulas employed in making the study are given in an appendix to the complete paper.

Net station load.....25,000 kw.
Load on auxiliary bus.....1,300 kw.
Gross station load.....26,300 kw.
Steam consumption of the main unit when carrying the gross station load
Case 1: 279,000 lb. per hr. Case 2: 270,000 lb. per hr.
High-pressure drips.....1000 lb. per hr.
Condensate losses.....1000 lb. per hr.
High-pressure steam losses.....3000 lb. per. hr.
Radiation losses from low-pressure steam:
2 per cent of total heat in steam used for heating the feedwater
Pressure of steam at throttle.....330 lb. per sq. in. gauge
Boiler pressure.....350 lb. per sq. in. gauge
Superheat.....200 deg. F.
Heat content of boiler steam.....1326 B.t.u. per lb.
Heat content of steam at throttle.....1324 B.t.u. per lb.
Vacuum on main unit.....29 in. Hg.
Temperature of condensate.....75 deg. F.
Temperature of make-up water entering evaporator.....60 deg. F.
Slope of Willans line of main unit.....Case 1: 12 lb. per kw-hr.
Case 2: 9 lb. per kw.-hr.
Heat content of the high-pressure drips recovered.....390 B.t.u. per lb.
Radiation, friction and generator losses of the house turbine in kw.:
650 kw. turbine, 65 kw. 1500-kw. turbine, 135 kw.
Radiation losses from bled steam on passing through the main unit:
1 per cent of load developed by the bled steam
Internal Rankine-cycle efficiency of bled steam based on steam pressure after throttle:.....Case 1: 67 per cent. Case 2: 80 per cent
Boiler efficiency when not using economizers.....78 per cent
Boiler efficiency when using economizers but not including the economizer efficiency.....75 per cent
Coefficient of heat transmission through economizers.....5 B.t.u. per sq. ft. per deg. mean temperature difference between flue gases and water
Temperature of gases entering economizers.....580 deg. F.
Specific heat of flue gases.....0.2375
Percentage of recoverable heat recovered by economizers.....85 per cent
Flue gases per lb. of coal.....19 lb.
Heating value of coal.....13,500 B.t.u. per lb.
Ratio of the load developed by the boilers to the full-load rating of the boilers when using economizers.....2.25

The Rankine-cycle efficiencies of the house turbines are given in Figs. 4 and 5. An average value of 80 per cent was used for the Rankine-cycle efficiency of the bled steam on the main unit, though this will vary 2 or 3 per cent either way, depending on the conditions of bleeding and the design of the turbine.

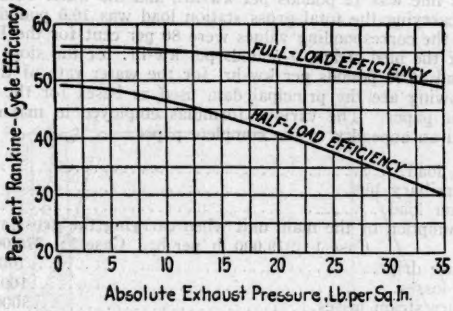

FIG. 4.—RANKINE-CYCLE EFFICIENCIES FOR CASE 1.

(Estimated Rankine-cycle efficiencies of 1500-1700 kw. house turbine generators as used for determining the heat balance for Case 1. The efficiencies used for the 650-kw. house turbine for Case 1 were approximately the same as those of Case 2, though the shape of its curve is similar to those given here. Steam presure, 330 pounds gauge; superheat, 200 degrees F.)

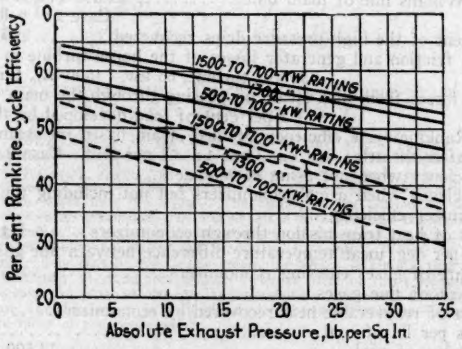

FIG. 5.—CASE 2. ESTIMATED RANKINE-CYCLE EFFICIENCIES OF TURBINE GENERATORS DESIGNED FOR ANY OF THE VARIOUS BACK PRESSURES AND OPERATING AGAINST THE BACK PRESSURE FOR WHICH THEY ARE DESIGNED.

(Steam pressure, 330 pounds gauge; superheat, 200 degrees F. The efficiencies would deviate somewhat from those shown with back pressures below 2 pounds absolute.)

All auxiliaries in these stations were considered as being motor-driven during normal operation. Surface condensers were used on the main units and the condensate before going to the feedwater heaters was passed as cooling water through the evaporator system supplying boiler-feed make-up water so long as this operated on exhaust steam. When the temperature of this exhaust steam was too low to evaporate the water efficiently and the use of live steam became necessary, the condensate was returned directly to the feedwater heater and the water used for the condenser of the evaporator was taken from the boiler feedwater previously heated in the economizers or in the feedwater heaters. With this arrangement the use of live steam on the evaporators did not materially affect the heat balance, and the pressure of the exhaust steam used for heating the feedwater was determined by the pressure within the feedwater heater rather than the requirements of the evaporator system. A schematic arrangement of these plants is shown in Fig. 6. Economizers are shown, but heat balances as well were worked out for plants not using economizers, in which case the boiler feedwater was delivered directly to the boiler.

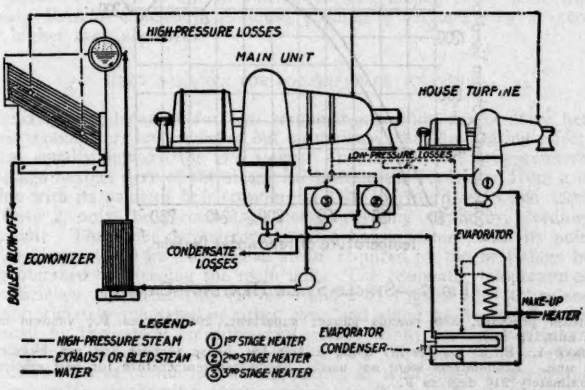

FIG. 6.—SCHEMATIC DIAGRAM OF STATION LAYOUT FOR THREE-STAGE FEEDWATER HEATING.

(The condensate from the main unit is circulated through the condenser of the make-up water evaporator system before going to the feedwater heater.)

When bleeding the main unit the pressure drops in the bleeder piping varied with the amount of steam bled, and this was taken into consideration. The boilers were operated between 175 and 200 per cent of rating when economizers were not used, and under this condition had an efficiency of 78 per cent. When economizers were used the operating capacity was increased to 225 per cent of rating and the efficiency reduced to 75 per cent, not including the economizers. The power taken by the induced-draft fans was not included in the auxiliary load, and to obtain the true heat consumption for the stations using economizers the equivalent heat consumption of these fans will have to be added to the rates given.

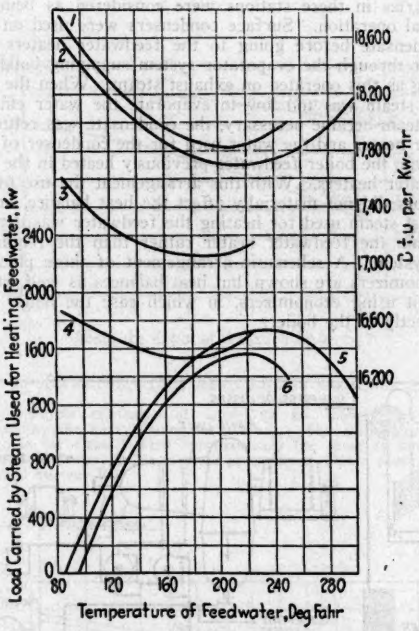

FIG. 7.—SINGLE-STAGE HEATING—CASE 1.

(Steam pressure, 330 pounds gauge; superheat, 200 degrees F.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1600-kw. house turbine and not bleeding the main unit. Economizers were not used. Feedwater temperature for best efficiency is approximately 210 degrees F.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit. Economizers were not used.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 square feet area and bleeding the main unit. Economizer surface approximately 50 per cent of the boiler area. The heat consumption of the induced draft fans, equivalent to the power consumed by them, not included.

CURVE 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 square feet of surface and bleeding the main unit. Though the economizer area is double that for Curve 3, the feedwater temperature for best economy is reduced only 10 to 15 degrees. Power taken by induced-draft fans not included with auxiliary power when determining heat-consumption curves.

CURVE 5. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit.

CURVE 6. Load developed by the 1600-kw. house turbine supplying exhaust steam to the feedwater heater. A comparison of the temperature at which the peak of Curve 6 occurs with that of Curve 5 indicates the influence that decreasing Rankine-cycle efficiencies with increasing back pressures on the house turbine have on the feedwater temperature for best economy. By reference to Curves 1 and 2, showing the B.t.u. rates per kw-hr., it is seen that the real influence of the decreasing Rankine-cycle efficiencies with increasing back pressures is small.

HEAT BALANCE FOR SINGLE-STAGE HEATING.

Heat balances for two arrangements using single-stage heating and no economizers were worked out. In one arrangement the auxiliary power was obtained from a house turbine of 650 kw. capacity, or one-half the total auxiliary load. The remainder of the auxiliary load was carried by the main unit. Steam, in addition to that supplied by the house turbine, in this case was bled from the main unit for heating the feedwater. The other arrangement used a house turbine with its point of best economy at 1600 kw. for Case 1 and 1700 kw. for Case 1, and was considered as being so designed that the house turbine could deliver power to the main bus. With this arrangement no means for bleeding the main unit were provided, all steam for heating feedwater being obtained from the house turbine. As indicated by the curves 1 and 2 of Figs. 7 and 8, the first arrangement is thermally the more economical and also requires for best efficiency a slightly higher feedwater temperature than the second arrangement. It is interesting to note that whereas the theoretical feed temperature for best efficiency was in the neighborhood of 250 degrees F., the actual temperature is closer to 200 degrees F. This is due to various factors, among them being the decreasing Rankine efficiency of house turbines when operating at successively higher back pressures.

HEAT BALANCE FOR DOUBLE-STAGE HEATING.

Similarly heat balances for two arrangements using double-stage heating and no economizers were worked out on the basis that the heating effect was divided equally between the two stages. In one of these arrangements the first-stage heaters derived the steam for heating the feedwater from a house turbine with its point of best economy at 1400 kw. for Case 1 and 1500 kw. for Case 2, only the second-stage heater deriving steam by bleeding the main unit. The other arrangement used a house turbine with its point of best economy at 650 kw., additional steam required by the first-stage heater being obtained by bleeding the main unit. The feedwater temperatures for best efficiency as indicated by curves 1 and 2 of Figs. 9 and 10 are seen to be approximately the same in either case, and about 25 degrees below that indicated by theoretical considerations alone. Both curves are rather flat over a range of 50 degrees in the vicinity of the point of best efficiency. The more efficient of the two arrangements, as with single-stage heating, is that one using the smaller house turbine, or the one bleeding the largest amount of steam from the main unit. This latter arrangement, with feedwater temperature of 275 degrees F., showed a possible saving of approximately 320 B.t.u. per kw-hr. as compared with single-stage heating. The maximum efficiency using single-stage heating, however, was obtained with a feedwater temperature approximately 200 degrees F. Comparing single-stage with double-stage heating on the basis that this temperature of 210 degrees F. was not to be exceeded, the difference between double-stage and single-stage heating is about 225 B.t.u. per kw-hr.

EFFECT OF ECONOMIZERS.

Economizers in connection with single- and double-stage heating were applied to those stations which used a 650-kw house turbine and bled the main unit this arrangement being the more economical. The economizers were assumed as having a heat-recovery factor of 85 per cent and a heat-transfer rate of 5 B.t.u. per hour per degree mean temperature difference

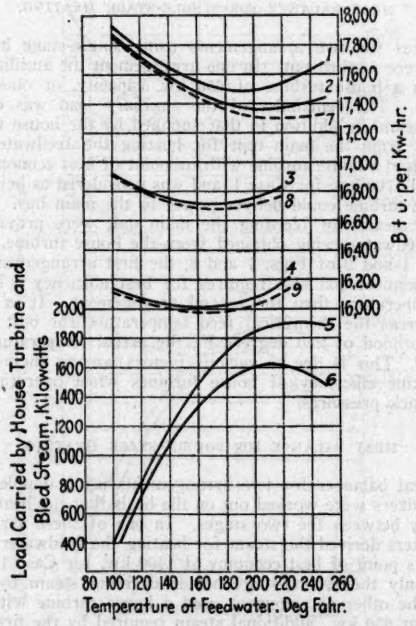

FIG. 8.—SINGLE-STAGE HEATING—CASE 2.

(Steam pressure, 330 pounds gauge; superheat, 200 degrees F.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1700-kw. house turbine and not bleeding the main unit. Economizers not used. The feed-water temperature for best economy is approximately 190 degrees F. This is 20 degrees lower than the best feedwater temperature for the corresponding conditions of Case 1, though due to the flatness of the curves over this range, the proper temperature for either case may be considered as approximately the same. The difference in temperature indicated is due to the difference in the slopes of the Rankine-cycle efficiency curves for the house turbine.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit not using economizers.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 square feet area, and bleeding the main unit. The feedwater temperature for best economy is lower than the corresponding temperature for Case 1, due largely to the difference between the slopes of the Willans lines of the main units. Power consumed by induced-draft fans not included in auxiliary power.

CURVE 4. B.t.u. per kw-hr. using a 650-kw, house turbine, economizer of 49,450 square feet of surface, and bleeding the main unit. Power consumed by induced-draft fan was not included in auxiliary power.

CURVE 5. Load developed by the steam used for heating the feedwater using a 650-kw. house turbine and bleeding the main unit.

CURVE 6. Load developed by the 1700-kw, house turbine supplying exhaust steam to the feedwater heater.

CURVE 7. Same as Curve 2 except that low-pressure steam losses are neglected.

CURVE 8. Same as Curve 3.

CURVE 9. Same as Curve 4 except that low-pressure steam losses are not included.

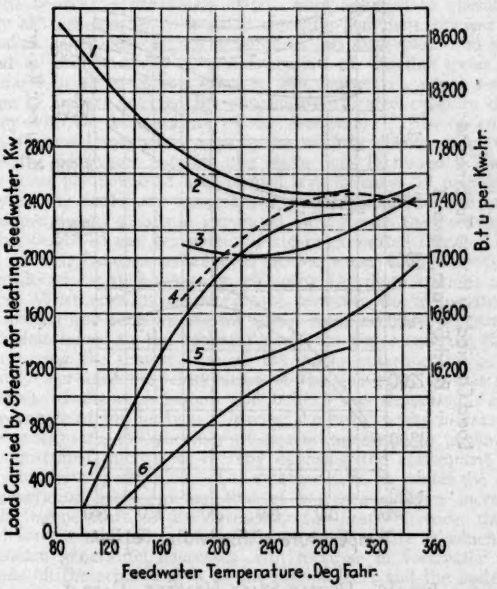

FIG. 9.—DOUBLE-STAGE HEATING—CASE 1.

(Steam pressure, 330 pounds gauge; superheat, 200 degrees F., vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1400-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately 280 degrees F.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. It is noticeable that as the number of stages used for heating the feedwater increases, the curve of heat consumption per kw-hr. flattens out.

CURVE 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 square feet area and bleeding the main unit at two points. Economizer surface approximately equivalent to 45 per cent of boiler area. Power taken by induced-draft fans not included in the auxiliary load.

CURVE 4 (dotted). Load developed by steam used for heating feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

CURVE 5. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 square feet area and bleeding the main unit at two points.

CURVE 7. Load developed by the steam used for heating the feedwater when using a 1400-kw. house turbine and bleeding the main unit only for the second-stage heating. Curve 6 shows the load carried by the house turbine for this arrangement of heating the feedwater.

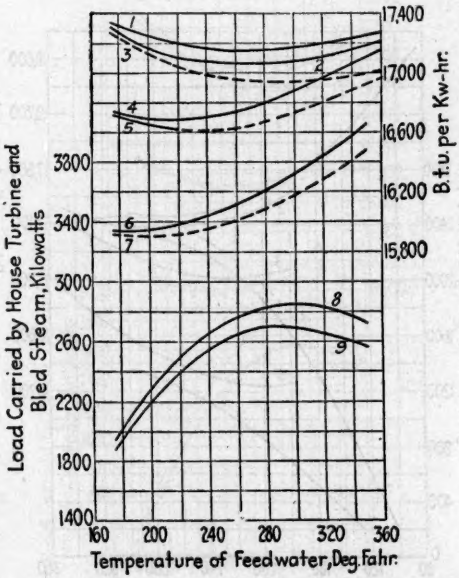

FIG. 10.—DOUBLE-STAGE HEATING—CASE 2.

(Steam pressure, 330 pounds gauge; superheat, 200 degrees F.; vacuum on the main unit, 29 in.)

CURVE 1. B.t.u. per kw-hr. using a 1500-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately the same as for the same conditions of Case 1.

CURVE 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. Curve 3 is the same except that losses due to leakage from the low-pressure steam piping are not included.

CURVE 4. B.t.u. per kw-hr, using a 650-kw. house turbine, economizer of 18,180 square feet area and bleeding the main unit at two points. The feedwater temperature for best economy is slightly lower than for Case 1. The power taken by the induced-draft fans was not included in the auxiliary power load. Curve 5 is the same, but does not include the low-pressure steam losses.

CURVE 6. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 square feet area and bleeding the main unit. The feedwater temperature for best economy is lower than for Case 1, due to difference in the slope of the Willans lines of the main units. Curve 7 is the same but does not include the low-pressure steam losses.

CURVE 8. Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

CURVE 9. Load developed by the 1500-kw. house turbine when used to supply steam for feed heating in conjunction with bleeding the main unit.

between the flue gases and the water. Two sizes of economizers were applied to each station to illustrate the effect of varying the size of the economizers. The boilers, as previously stated, were assumed to operate at approximately 225 per cent of their rated capacity, and their efficiency in this case was taken as 75 per cent, or 3 per cent less than that used when the stations had no economizers. The temperature of the flue gases entering the economizers in all cases was taken as 580 degrees F., which meant that for the type of boilers selected the percentage of rated capacity developed did not vary with the temperature of the feedwater. This, of course, required that the total operating capacity be slightly decreased as the temperature of the feedwater entering the boiler was increased. The weight of the flue gases per pound of coal burned was taken as 19 pounds and independently of the feedwater temperature, but, inasmuch as the weight of coal burned per pound of steam generated varied with the feedwater temperature, the weight of gas per pound of steam generated likewise varied.

Analysis of heat balances as affected by feedwater temperatures did not involve considerations of boiler-room efficiency for those stations not using economizers. When economizers were used, however, the temperature of the flue gases leaving the economizer, for given equipment and operating conditions, was determined by the temperature of the water entering the economizer. Increasing the temperature of the water entering the economizer simultaneously increased the temperature of the flue gases so that the combined efficiency of the economizers and boilers was decreased. However, as the temperature of the feedwater leaving the heaters was increased above that of the condensate, the efficiency of converting steam to power, as previously demonstrated, increased, thereby opposing the consequent decrease in efficiency of steam generation. The relative rates at which the efficiency of steam generation decreased and that of power generation increased determined the temperature of the feedwater for best efficiency, these rates being equal for the condition of best efficiency. The rate at which the efficiency of steam generation decreases with increase in feedwater temperatures depends on the relative area of the economizers and the boilers. The larger the economizer relative to the boiler, the more rapid is the rate at which this efficiency falls off. In consequence of this, the feedwater temperatures for best efficiency were found to be lower for the larger economizers than for the smaller ones, though the differences were not of great moment.

The temperature for best economy when economizers are used, as given by the heat-consumption curves 3 and 4 of Figs. 7 and 8 for single-stage heating, lies between 175 and 190 degrees F. for Case 1 and 165 and 175 degrees F. for Case 2. With double-stage heatings, Figs. 9 and 10, the corresponding temperatures are 225 and 175 degrees F. for the smaller and larger economizer surfaces, respectively, for Case 2, and 230 and 200 degrees F. for Case 1. The temperatures for Case 1 are lower than those for Case 2, due to the difference in the slopes of the Willans lines used. For each kilowatt-hour developed by the steam used for heating the feedwater in Case 1, the steam condensed in the main unit's condenser was reduced 12 pounds, while for Case 2 this figure was 9 pounds. The benefit derived from carrying load on steam used for heating the feedwater was therefore relatively less for Case 2 than for Case 1, while the economizer and boiler efficiency remained the same. The differences in temperature are small, however, and indicate that with fair accuracy the desirable feedwater temperature may be considered as a range which is largely independent of the characteristics of the steam equipment used. The economizers used for double-stage heating were slightly smaller than those used for single-stage heating, commercial considerations indicating that this was justified. The

difference in the sizes of the economizers did not, however, materially affect the feedwater temperature for best efficiency. The rate-of-heat-consumption curves are rather flat over a considerable range in proximity to the temperature for best economy, and increasing the area of the economizers even to the extent of doubling them need not require large changes in the temperature of the feedwater. The various data used in connection with the economizers are given in curves in the complete paper.

COMPARISON OF RESULTS WITH AND WITHOUT ECONOMIZERS.

A comparison of feedwater temperatures for best economy as indicated by Case 1 and Case 2, respectively, shows that when economizers are not used ordinary variations in the Rankine-cycle efficiency of the bled steam, the efficiency curve for the house turbine, and the slope of the Willans line of the main unit have no considerable effect. When economizers are used, the temperatures are reduced by decreasing the slope of the Willans line of the main unit, but, as previously stated, the influence is not large. It is evident that for the purpose of establishing the proper feedwater temperature for best economy minute accuracy in the determination of the various turbine efficiencies is not required. The overall efficiency of the entire station is influenced, of course, by these efficiencies, but the best feedwater temperature changes only slightly with them. For this study a constant efficiency for bled steam was used. However, the house-turbine efficiencies were considered as being a function of the feedwater temperature, and a comparison of the temperatures for best economy obtained when using a house turbine alone with those obtained when using a smaller house turbine together with bleeding the main unit illustrates the influence of the variation in the Rankine-cycle efficiency. The difference in the temperatures that were obtained is between 10 and 15 degrees F., but inasmuch as the curves are flat this is of no great consequence.

MULTIPLE-STAGE HEATING.

By similar methods of computation, heat balances for four-stage heaters with and without the use of economizers were determined. The average Rankine-cycle efficiency for the bled steam in this case was taken at 79 per cent instead of 80 per cent, as the average efficiency over four stages would probably be somewhat less than that for one stage. Fig. 11 shows the heat consumption per kilowatt-hour for the various methods of heating the feedwater worked out for Case 2, using the 650-kw. house turbine. As the number of heating effects increase, the value of the last effect decreases, which is to be expected. It is interesting to observe also that as the number of effects increase, the heat-consumption curves for the various stations flatten out in proximity to the temperature for best economy and that it would therefore seem undesirable to go beyond a certain feedwater temperature regardless of the number of stages. When economizers of 21,850 square feet are used the gains in efficiency referred to a basis of no heating for single-, double-, triple-, and quadruple-stage heating are 1.86, 2.88, 3.35, and 3.64 per cent, respectively. When the number of heating effects is increased the value of the last heating decreases, and in determining the proper number of stages consideration must be given to the investment and also to the operating problems encountered.

AIR ECONOMIZERS.

If air for the boilers is preheated by means of air economizers and no feedwater economizers are used, such preheating does not affect the feedwater temperature for best economy. However, the air may be preheated by exhaust steam similarly to heating feedwater. Also air economizers and

exhaust-steam air heating may be used with or without the coincident use of water economizers, and with single-or multiple-stage feedwater heating. If air is heated by exhaust steam alone, the problem of determining its temperature for maximum efficiency is not different from that for determining the temperature of feedwater heated only by exhaust steam. This, however, is not the case if both air and water economizers are used simultaneously.

Air and water economizers, if used simultaneously, may be placed either in parallel or in series. If placed in parallel the effect of the air economizer is to reduce the amount of gases available for heating the feedwater, and so alter the operating characteristics of the water economizers so far as these

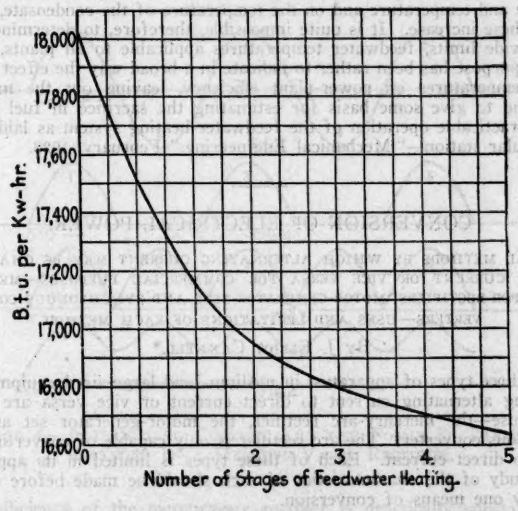

FIG. 11.—HEAT CONSUMPTION PER KW-HR. FOR VARIOUS STAGES OF FEEDWATER HEATING—CASE 2.

are determined by the ratio of the waste gases to the water heated. If the air heater is placed in series with the water economizer and in the coldest part of the flue gases, the leaving temperature of these gases is no longer determined by the temperature of the boiler feedwater. For such an arrangement it would appear desirable to heat the boiler feedwater to that feedwater temperature giving the maximum efficiency for the conversion of steam energy to electrical power. The feedwater would then be further heated in a water economizer while the temperature of the flue gases leaving the water economizer could be reduced to a desirable point, justified, of course, by the efficiency of the air economizers and their cost. On this basis the temperature of the feedwater entering the economizers would correspond to that previously determined when no economizers were used. The air economizers in this case, of course, would have to reduce the temperature of the flue gases below that established for the use of water economizers alone.

CONCLUSIONS.

So many factors enter into the determination of the proper feedwater temperature of a plant that figures determined for one station should not be directly applied to another. As seen, the temperature should be lower for plants using single-stage heating than for those using multiple-stage heating, and also should be lower for plants using economizers than for plants not using economizers. Efficiencies of auxiliary apparatus as well as of the main generating units have their influence. If the feedwater temperature be raised much above 212 degrees F., factors incident to the use of pressures well above atmosphere in the auxiliary exhaust piping come into play. The proper feedwater temperature is dependent somewhat upon the initial steam pressure and temperature and on the temperature of the condensate, increasing as these increase. It is quite impossible, therefore, to determine, except within wide limits, feedwater temperatures applicable to all plants, and the present purpose has been rather to indicate in a broad way the effect of feedwater temperatures on power-plant efficiency, leaving out the matter of costs, and to give some basis for estimating the sacrifice in fuel made to assure practicable operation of the feedwater-heating system as laid out for a particular station—"Mechanical Engineering," February, 1923.

CONVERSION OF ELECTRICAL POWER.

PRACTICAL METHODS BY WHICH ALTERNATING CURRENT MAY BE CHANGED TO DIRECT CURRENT OR VICE VERSA FOR COMMERCIAL PURPOSES—MERCURY VAPOR RECTIFIERS, MOTOR-GENERATOR SETS AND SYNCHRONOUS CONVERTERS—USES AND LIMITATIONS OF EACH METHOD.

By J. Elliot Cannell.*

Only three types of apparatus, in medium- and large-sized equipment, for converting alternating current to direct current or vice versa are now in general use—the mercury-arc rectifier, the motor-generator set and the synchronous converter. The arc rectifier is only capable of converting alternating to direct current. Each of these types is limited in its application, and a study of the characteristics of each should be made before deciding upon any one means of conversion.

MERCURY-ARC RECTIFIERS.

The chief applications in this country of the mercury-arc rectifier are for charging storage batteries and for operating lamps on series arc-light circuits; in Europe they have been applied commercially to industrial and railway loads. The rectifier consists principally of a glass bulb containing ionized mercury vapor at low pressure (the air having been exhausted), having one electrode of mercury and two or more electrodes of some other conductor, such as iron. A high resistance is offered to a current flowing from the mercury to the iron electrodes, but a low resistance is obtained for a current flowing in the opposite direction. The current flow through the mercury vapor is started by forming an arc in the tube between the mercury electrode and an auxiliary starting electrode, by tilting the tube, Fig. 5.

If only one iron electrode were used, the current between the iron and mercury electrodes would have such a wave form as in Fig. 2, one-half of the alternating-current wave, Fig. 1, being suppressed, since the current

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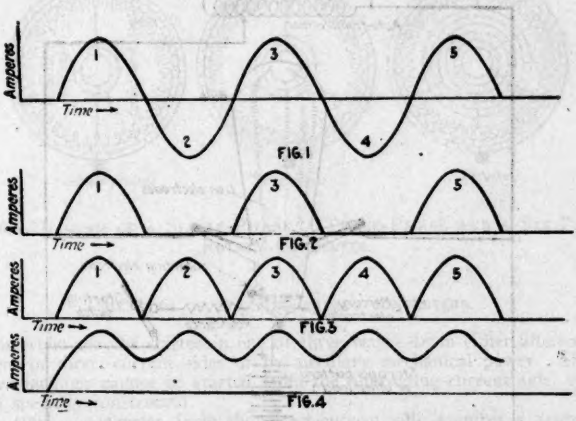
flows in only one direction. But if two iron electrodes be used, every alternation of the current will be rectified, and the wave form will be as in Fig. 3.

If the current between the electrodes drops to a zero value, as is the case in Figs. 2 and 3, the mercury vapor loses its ionization and the tube will have to be started over again. To avoid this, an inductance is inserted in the tube circuit, and the current wave then takes the form of Fig. 4, being a varying direct current. For the successful operation of a rectifier, an auto-transformer, two inductances and a resistance are also needed, as shown in Fig. 5.



FIGS. 1 TO 4.—ALTERNATING CURRENT AND THE FORMS IT TAKES WHEN RECTIFIED IN A MERCURY-VAPOR RECTIFIER.

The efficiency of the mercury-arc rectifier is practically constant at all loads, but varies with the voltage, being high at high voltages, ranging anywhere from about 94 per cent down and operates at a power factor as high as 98 per cent in the large steel-clad types. The useful life of the tube is generally from 400 to 1,000 hours, and tube renewals cost from 5 per cent to 10 per cent of the first cost of the outfit.

MOTOR-GENERATOR SETS.

Motor-generator sets are made to convert alternating current to direct current or vice versa, also to transform direct-current power of one voltage to direct-current power of another voltage and to change alternating current of one frequency and voltage to alternating-current power of another frequency and voltage.

Motor generators are used to convert alternating current commercially where a wide variation in direct-current voltage is necessary or where it is desirable to have the alternating-current and direct-current sides independent. When so used, the motor generator usually consists of an alternating-current motor of either the synchronous or the induction type directly connected to a direct-current generator. The synchronous-motor drive is more

generally used because of its higher efficiency and the fact that it can be made to take a leading current, thus bettering the power-factor condition on the supply lines.

A motor generator is less efficient than the synchronous converter because of a double transformation of energy. The over-all efficiency of a motor-generator set is the product of the efficiencies of the motor and generator and ranges between 80 and 92 per cent.

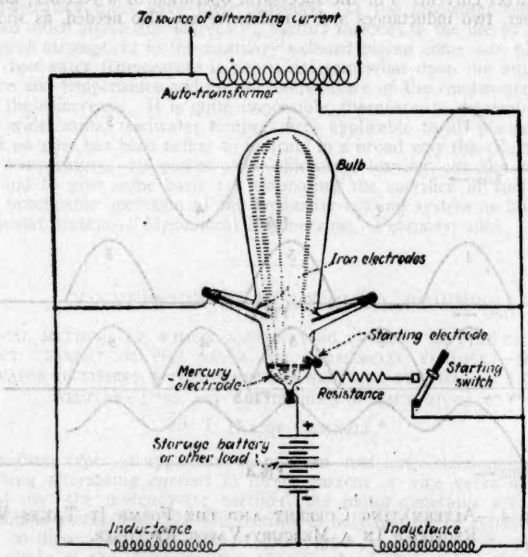


FIG. 5.—DIAGRAM OF MERCURY-VAPOR RECTIFIER.

SYNCHRONOUS CONVERTERS.

A synchronous converter resembles a direct-current generator and, from outward appearances, differs from the latter only in having slip rings at one end of its armature and a commutator at the other end. By supplying, alternating-current power to the slip rings, direct-current power may be taken from the commutator or vice versa. When used to convert alternating current into direct current the converter is said to be direct, and when used to convert direct current into alternating current it is said to be inverted. When used inverted, the converter requires speed-limiting devices to protect it against excessive speeds caused by accidental weakening of the field strength. The converter is, however, generally used direct.

The armature winding of a converter is the same as a direct-current generator having taps from points equidistant on the windings connected to the slip rings. The number of taps is the same as the number of phases of the alternating current, on two-pole machines, except for single-phase current, which requires two taps.

Schematic connections are shown in Fig. 6 for a single-, three- and a six-phase rotary converter. A machine having an armature connected in this manner may be used as a direct-current generator or motor, as an alternating-current generator or synchronous motor, as a direct or inverted converter, and as an alternating-current and direct-current generator simultaneously by having its armature driven mechanically. Any direct-current generator provided with suitable taps would theoretically act as a converter, but its design and proportions might make its operation unsatisfactory.

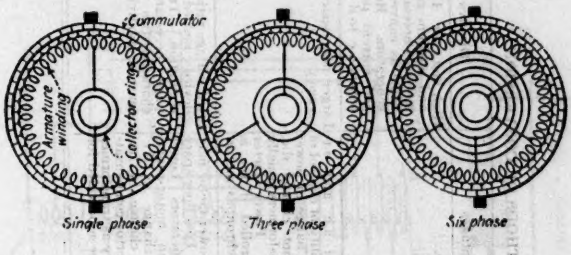


FIG. 6.—DIAGRAM OF A SINGLE-PHASE, A THREE-PHASE AND A SIX-PHASE ROTARY CONVERTER.

METHODS OF STARTING ROTARY CONVERTERS.

Converters may be started in one of three ways—from either alternating-current or direct-current sides or by auxiliary mechanical power. Single-phase machines cannot be started from the alternating-current side, except when specially constructed.

To start a converter from the direct-current side requires a source of direct-current power, which is usually not available unless there are other converters in operation in the same station. With this method of starting, the machine is brought up to speed by cutting out resistance in the armature circuit, and it is then synchronized with the alternating-current supply.

In starting by mechanical power, an induction motor is usually employed to bring the armature of the converter up to speed. The latter is then synchronized with the alternating-current supply. The induction motor should have one pair of poles less than the converter so that synchronous speed of the converter may be reached.

The most common method of starting a polyphase converter is by direct use of alternating power impressed on the alternating-current side, thus eliminating synchronizing. The alternating-current power can be reduced for starting purposes by using the mid-taps on the transformer secondaries. If this method of starting is to be used, a damping winding is built into the pole faces and the converter is started as an induction motor, the field being applied only after the armature is up to speed. A precaution necessary with this means of starting is to have the field subdivided into sections so that a dangerously high voltage will not be induced in it by transformer action, while coming up to speed.

Fig. 7 shows the connections for a six-phase converter, using diametral connections of the secondary leads. The reduced voltage for starting is obtained by mid-taps on the transformer secondaries. Reactances for voltage

COMPARISON OF THE THREE COMMERCIAL METHODS OF CONVERSION

	Uses	Limits	Advantages	Disadvantages	Efficiency, Per Cent
Mercury-arc rectifier	Charging storage batteries and operating arc lights. Railway work.	50 amperes capacity with glass tubes, 200 to 900 amperes with steel-clad tubes, 7,000 volts direct current for series arc lights.	Small floor space.	High first cost. Limited to small and medium capacity. Tube renewals necessary. Not as reliable as converters or motor-generators. Rectified voltage and current pulsating. Will rectify a c to d c only	From 98 down, depending on voltage
Motor-generator sets.	Convert a c. of any voltage to d.c. of any voltage and vice versa. Transforms d.c. of one voltage to d.c. of any other voltage. Change a.c. of one frequency to a.c. of another frequency or voltage.	Limited only in voltage and capacity as are motors and generators now on the market.	Independence of a.c and d.c. power. Flexibility of control of d.c. power. Change in power factor does not change d.c. voltage. Transformers usually unnecessary. Power factor may be controlled if set is driven by synchronous motor	Larger floor space. Lower efficiency	80 to 92
Synchronous converter	Railway work particularly, although may be applied to other uses if electrical conditions warrant.	1,500 volt d c. maximum. Made in capacity as large as 7,500 kw. No limit to a.c. voltage as transformers are necessary.	High-efficiency power factor may be controlled. Less floor space than motor-generators. Lower first cost than motor-generators. No change in d c voltage with change in frequency. Starter or compensator not necessary as transformer winding may be subdivided for this purpose.	Fixed voltage ratio. An auxiliary device needed for voltage regulation. Transformers always necessary. Starting characteristics poor. Field must be subdivided	85 to 98

adjustment between the transformers and collector rings are not shown, as most manufacturers wind transformers for use with converters with sufficient inherent reactance to give the desired degree of compounding of the direct voltage.

Synchronous converters may be operated successfully in parallel by the use of equalizers as with direct-current compound generators. When operated in parallel, each converter should be fed from a separate group of transformers; or separate secondaries for each machine excited by a common primary may be used.

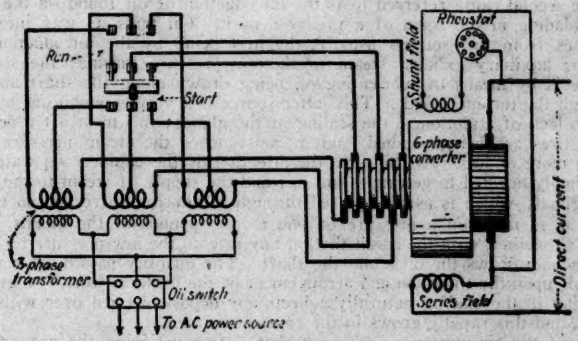


FIG. 7.—SIMPLIFIED WIRING DIAGRAM OF ROTARY CONVERTER.

Reverse-current relays should be provided on the direct-current circuit breakers, and it is usually desirable to interconnect both alternating-current and direct-current circuit breakers so that in case one opens the other will also open.—"Power," Jan. 16, 1923.

SOME USEFUL NOTES ON THE CLEANING OF MARINE TURBINES.

It is often found, from the record of pressures taken, that silting up or choking is taking place in the blading of marine turbines; consequently, the efficiency of the machinery falls off rapidly as the amount of choking increases. When this condition is noted, it may not be convenient or even possible to open up the turbines for hand cleaning.

The principal causes of choking in the turbines are, first, sediment carried over with the steam from the boilers, and secondly, grease or oil becoming introduced. Very often the effect is due to a combination of the two causes.

Dealing with the first, it is to be remembered that with turbine machinery vacuum is important and at all times should be maintained at the highest for which the turbine is designed. The demand on the boilers for steam is constant, and, owing to the vacuum, insistent; consequently, there is considerable inducement for priming to take place, and water and sediment to be carried over with the steam. This sediment, or at least some considerable portion of it, inevitably adheres to the turbine blading.

Many forms of mechanical separators or steam purifiers have been introduced to cope with this trouble, some of them have proved very effective in service, and have reduced, almost to vanishing point, the amount of water and sediment carried over from the boilers.

The composition and quantity of the sediment in the boilers will, however, vary with the water used and the treatment adopted, but in all cases there is, in the sediment formed, a considerable amount of light, finely divided or flocculent matter which floats on the surface of the water and is picked up with the steam and carried over in spite of the mechanical means used to intercept it.

The second cause referred to is the introduction of oil found on the turbine blading in the form of a greasy deposit. Oil finds its way into the turbines from two sources usually, the first being by the introduction of surplus auxiliary exhaust steam to the low-pressure turbine; the second source is by means of lubricating oil being drawn along the shaft and in through the turbine glands. This latter source is a very common one and is due to lack of attention to the sealing of the glands; this most often occurs when fires are being cleaned, and in consequence the steam pressure has fallen more or less. At this time the attention of the engineer on watch is principally devoted to getting steam up; and the supply of steam to the turbine glands, which is usually passed through a reducing valve and so regulated as to maintain a pressure of one to two pounds at the glands, may cease altogether, with the result that, at any rate on the low-pressure turbine, the vacuum draws the oil along the shaft. The quantity passed, of course, depends upon the condition and arrangement of the baffles fitted. The grease adhering to the blading naturally collects any deposit carried over with the steam, and this rapidly grows in the course of time.

Where the circumstances are such that (after verifying the correctness of the pressure gauges and being satisfied that the blading is choked) it is not possible to open up the turbines for cleaning, satisfactory results can be obtained by the application or introduction of turpentine and Atlas preservative.

On the day of arrival in port, after the ship is moored and before the machinery is finally shut down, shut off or blank the discharge of the air pump to the feed-filter tank and arrange for the pump to discharge to the bilges. Or, if this time is not a suitable one for the operation, the turbines should be heated up, say, on the day before sailing, and the air pump, as already stated, arranged to discharge to the bilge. Run the turbines under steam slowly, say 10 to 15 R.P.M. of the propeller shaft, having a vacuum of 15 to 20 inches at the condenser.

Heat up the turpentine to about 100 degrees, connect a small pipe, such as is used for pressure-gauge connections, to a gauge cock on the high-presture turbine as near the inlet end as the effect of the vacuum is obtainable. Allow half the quantity of turpentine, mentioned hereunder, to be *slowly* drawn into the revolving turbine. After this quantity has passed into the high-pressure turbine, pass the remainder of the turpentine in a similar manner into the ahead inlet end of the low-pressure turbine, or after end of the intermediate-pressure turbine, where three turbines are fitted in series.

After the total quantity of turpentine has been introduced as described, have heated up two-thirds of the quantity of Atlas preservative and pass this into the early stage of the high-pressure turbine at full strength. The remaining quantity of preservative should be diluted with five times its quantity of boiling water, and be allowed to follow through.

It must be emphasized that the liquids should be drawn into the turbines very slowly.

Samples of the air pump discharge should be collected from time to time for inspection.

Quantities recommended for ordinary cases:—

Single-Reduction Geared Turbines.—Turpentine: 2 gallons per 1,000 S.H.P.; Atlas preservative: 6 gallons per 1,000 S.H.P.

Double-Reduction Geared Turbines.—Turpentine: 1½ gallons per 1,000 S.H.P.; Atlas preservative: 4½ gallons per 1,000 S.H.P.—"The Marine Engineer and Naval Architect," March, 1923.

CARBON DIOXIDE VERSUS AMMONIA AS A REFRIGERANT.

By H. J. Macintire.

ASSOCIATE PROFESSOR OF REFRIGERATION, UNIVERSITY OF ILLINOIS.

The refrigerating machine using carbon dioxide came into use about forty years ago (in 1882) and therefore only about six years after the first successful ammonia compression machine was devised by Carl Linde. Yet since then the ammonia machine in this country has become strongly established, whereas carbon dioxide as a refrigerating medium has had a hard fight and a successful one only in certain classes of work. There must be a particular reason for this partiality which is worth while going into detail.

It is frequently said that the pressure range encountered in the use of carbon dioxide is the reason for the preference of ammonia, but this is not conclusive. The fact that pressures of 800 to 1,000 pounds per square inch or over are met with is not insurmountable. At the present time steam pressures of 350 pounds are common and 500 to 600 pounds are beginning to be used. The existence of a gas pressure is not serious in these days when good quality steel and wrought-iron pipe is available and the flanged joint can be cheaply produced and made tight. Also the use of welded pipe is doing away with all except a few of the flanged joints.

With regard to the action of carbon dioxide in the compressor, except for the fact that heavy unit pressures are carried and therefore that the cylinder has to be constructed satisfactorily for this purpose, the machine has a simpler problem than many ammonia compressors in daily use. In gas compression there is interest primarily in the ratio of the suction and the discharge pressures, or $P_2 \div P_1$. The temperature of the discharged gas as well as the work done on the gas is in proportion to some function of this ratio of pressures. As a matter of fact, the dry-vacuum air pump for steam-turbine use, which discharges into the atmosphere, has a much more severe duty, as has also the air compressor discharging compressed air at 100 pounds gauge, than has the carbon dioxide compressor operating at 0 degree F. (303 pounds absolute) and with a condenser pressure of 1,000 pounds. In the first case the ratio of the discharge pressure to the suction pressure is approximately $15 \div 0.5 = 30$; in the second the ratio is $115 \div 15 = 7\frac{2}{3}$, and in the third case the ratio is $1{,}000 \div 303 = 3.3$.

SIZE OF COMPRESSOR.

From the foregoing it is questionable whether the pressures alone existing in the refrigerating cycle exert a deciding influence on the selection of any refrigerant except in particular cases. There must be some other reason for a preference. This may be as regards the size of the compressor per ton of refrigeration. The carbon-dioxide compressor has the advantage, however, in this case even with warm condensing water. At 0 degrees F. boiling temperature of the refrigerant and with 70 degrees F. condensing

water the carbon-dioxide compressor requires about 1,500 cubic inches piston displacement per ton of refrigeration per minute and the ammonia compressor for the same conditions requires about 8,000 cubic inches. The carbon-dioxide compressor, therefore, has a decided advantage as far as the piston displacement is concerned for nominal temperatures of cooling water and boiling temperatures of the refrigerant. But this advantage is not uniform.

It is well known that carbon dioxide has a low critical temperature. At and above 88.4 degrees F. the carbon-dioxide vapor cannot be condensed no matter what the pressure on the vapor may be. Should the high-temperature condenser water only be at hand—as, for example, cooling-tower or spray-nozzle water at 90 degrees F.—then the pressure in an ammonia plant would be about 200 pounds gauge, in the carbon-dioxide plant about 1,400 pounds. The carbon dioxide would not condense in the condenser as we understand it, for it will not condense at 88.4 degrees or at any point above the critical temperature. The mistake must not be made that it ceases to act as a volatile liquid, however. Some engineers believe that carbon dioxide can give no refrigeration at all with condenser temperatures above 88.4 degrees F., and others claim that the refrigeration is slight and comparable with what would be possible with a "dense gas," as air for example. Neither of these ideas is correct. The so-called critical temperature is where the "latent heat" of liquefaction is zero, but the vapor above this temperature is dense and behaves much like a liquid. On passage through the expansion valve, the uncondensed gas becomes part liquid and this liquid is the means for the useful refrigeration obtainable.

There is this disadvantage, however: The latent heat for carbon dioxide is much smaller than that for ammonia, being from 130 to 115 B.t.u. per pound for ordinary boiling temperatures, whereas ammonia is from 597 to 569 B.t.u. In spite of the fact that the specific heat of ammonia is greater than unity and that for carbon dioxide it is about 0.33, the proportion of useful refrigeration is less at usual refrigerating conditions than for ammonia. For example, the ratio of possible refrigeration at 50 degrees F. and at 80 degrees F. condensing temperature in each case, with 0 degree F. boiling temperature, is for ammonia 0.934 and for carbon dioxide, 0.795. This loss for carbon dioxide becomes even greater with cooling water of higher temperature in the condenser, but it may be said that the usual loss of capacity is that shown with an 80-degree condenser condensate, representing a fairly liberal amount of 70 degrees F. cooling water. However, this does not show the whole story, as will now be explained.

In the table the boiling temperature in the evaporating coils is taken as 0 degrees F.

AMMONIA VERSUS CARBON DIOXIDE.

	Net Refrigeration per Pound, B.t.u.				Ratio of Capacity With 80 Deg. Liquefaction To That at 50 Degrees.
Liquefaction temperature.	50	60	70	80	.....
Ammonia	13.7	502.4	491.1	479.6	0.934
Carbon dioxide	89.2	83.1	77.0	70.9	0.795

Using 70 degrees F. cooling water and the resulting 80 degrees condensate with 0 degrees boiling temperature of the refrigerant, the net theoretical refrigerating effect of the two refrigerants are:

611.6 — 132.0 = 479.6 *B.t.u. per pound for ammonia.*
99.8 — 28.9 = 70.9 *B.t.u. per pound for carbon dioxide.*

If the liquid refrigerant could go to the expansion valve at 0 degrees F., the temperature at which refrigeration in this assumed case takes place, then there would be 568.7 B.t.u. per pound for ammonia and 117.5 B.t.u. per pound for carbon dioxide. The ratio of useful refrigeration for ammonia would be 479.6 ÷ 568.7 = 0.844, and for carbon dioxide, 70.9 ÷ 117.5 = 0.604. In other words, because of the fact that the condenser usually cools the condensate to a temperature some 5 to 10 degrees or more above the initial temperature of the cooling water, the refrigerant has to cool itself at the expansion valve to the evaporating temperature before useful refrigeration can be performed. The 80 degrees liquid, then, has to be reduced to 0 degrees F., and in the performance of this duty some liquid is evaporated and this becomes useless for further refrigeration. This loss is, for the case stated, about 15 per cent for ammonia and 40 per cent for carbon dioxide.

CONDENSER TEMPERATURES.

The crux of the whole matter is that carbon dioxide is a cool condensing-water refrigerant not to be preferred for use in the case of hot sea water, tropical surface water or for the usual cooling-tower installations where the water available for the condenser rises in temperature in the condenser to a point near or above the critical temperature for carbon dioxide of 88.4 degrees F. Carbon dioxide suffers also with a wide range in the operating conditions as in the case of the 0 degrees boiling temperature and the 80 degrees condensate example given previously. This is because the latent heat of vaporization of carbon dioxide is small as compared with ammonia and because a relatively larger amount of possible refrigeration is consumed in cooling the liquid. This amount of liquid boiled in cooling the liquid from the condenser is returned to the compressor and has to be compressed with the vapor which has performed useful refrigeration. The horsepower per ton of refrigeration and the water used in the condenser are influenced directly by these losses, and it should be expected that both the horsepower per ton and the amount of water used for any particular set of conditions would be greater than would be required for ammonia.

ADVANTAGES OF AMMONIA.

Figs. 1 to 6 inclusive show better than has ever been presented before, the relative advantages of ammonia. The brake-horsepower per ton for similar operating conditions is about 10 to 25 per cent greater for carbon dioxide than for ammonia. In each case the mechanical efficiency is assumed to be 90 per cent and allowance is made for the volumetric efficiency of the compressor. The piston displacement per ton per minute is, however, much smaller for carbon dioxide than for any other refrigerant, although, because of the heavy unit compressor pressures and the resulting desire to decrease the cylinder bore, but to compensate for the same by a long stroke as compared with the bore, the compressor stroke and therefore the frame, the flywheel and the condenser are of the same proportions or larger than for ammonia.

These are the reasons why carbon-dioxide compressors are not adaptable in the United States for ice making except as part of a hotel or hospital plant, or for cold-storage work or in the packing-house and chemical industries, where the cost of a ton of refrigeration per hour or per 24 hours is of sufficient importance to be considered above other factors. In ice making the primary consideration where competition is at all keen is whether 40, 50 or 60 kw.-hr. are required per ton of ice. If carbon dioxide is used as a refrigerant, then, it must be for reasons more pressing than those already

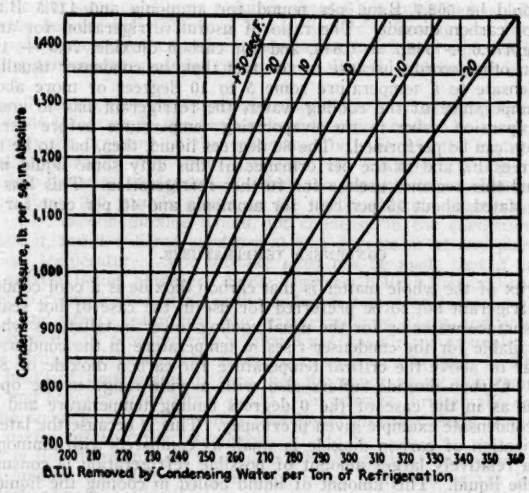


FIG. 1.—HEAT TRANSFER IN CO_2 CONDENSER PER TON OF REFRIGERATION.

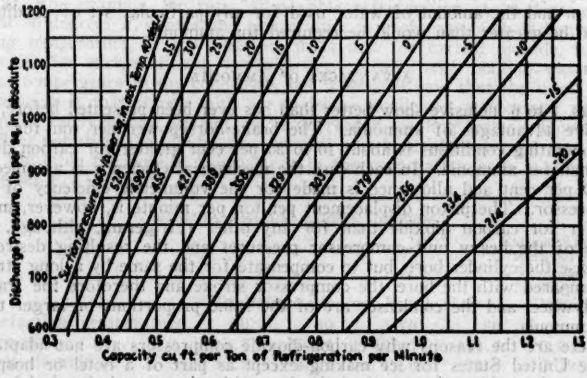


FIG. 2.—VOLUME OF CO_2 PER TON OF REFRIGERATION.

No liquid cooling; dry compression; allowance made for real volumetric efficiency.

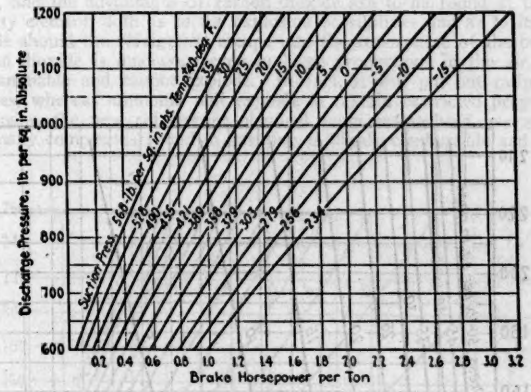


FIG. 3.—BRAKE HORSEPOWER PER TON OF REFRIGERATION FOR CO_2.

Dry compression; no clearance; b.hp. = i.hp. + 10 per cent.

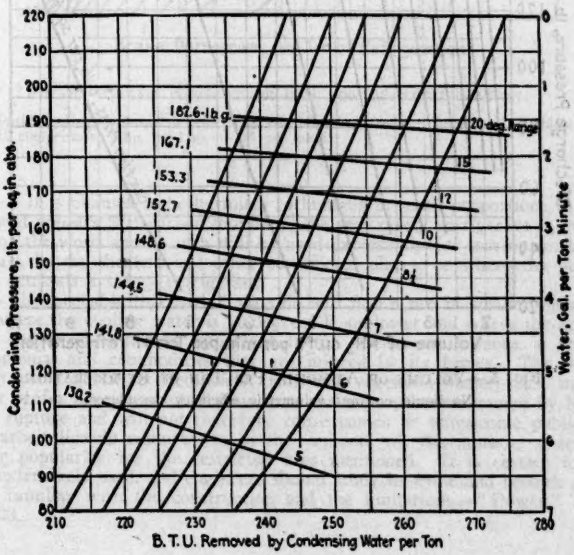


FIG. 4.—B.T.U. REMOVED PER TON IN AMMONIA CONDENSER.

No liquid cooling; 70-degrees F. initial temperature cooling water.

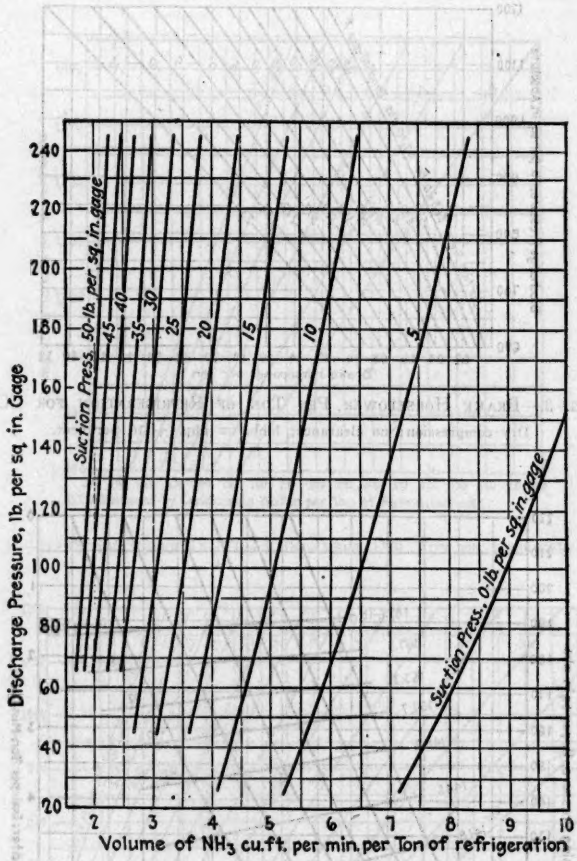


FIG. 5.—VOLUME OF AMMONIA PER TON OF REFRIGERATION.
No liquid cooling; volumetric efficiency considered.

outlined, and the advantages of carbon dioxide are to be found at times in the safety element, both as to the explosion possibilities and as to its effect on people should the refrigerant escape into the atmosphere of the building.

Carbon dioxide is innocuous even in large proportions in the air. It is non-inflammable and cannot explode. In fact, it will put out rather than start fires, whereas ammonia will explode in certain restricted proportions, and propane, the new refrigerant obtained from casing-head gas and one of the many compounds found in gasoline, is highly combustible and an ex-

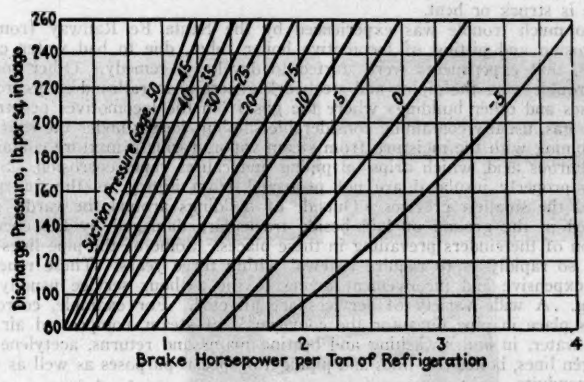


FIG. 6.—BRAKE HORSEPOWER PER TON OF REFRIGERATION.

No liquid cooling; volumetric efficiency considered; based on horizontal double-acting ammonia compressor; b.hp. = i.hp. + 10 per cent.

plosive. It is claimed to be harmless to humans in small proportions in the air, but it appears that carbon dioxide is the real safety refrigerant in every sense of the word and as such will be used extensively for marine work, in hospitals, hotels, theaters and other constricted places, or where the public safety demands a safety refrigerant.

It is also probable that carbon dioxide will find a use in smaller installations where the cooling water is 70 degrees F. or lower and where the actual power consumed is not an important factor. There are persons in terror of ammonia, and commodities that are injured by its fumes. The small market man, florist, confectioner, fruit and vegetable dealer, etc., might readily prefer a refrigerant that will not make its presence known by leaks or by rupture and will not therefore cause panics or unwelcome publicity. The carbon-dioxide compressor, in the opinion of the author, deserves greater popularity for the restricted uses mentioned. It is certain to be more extensively used, and engineers should study its cycle and become generally familiar with the construction and the limitations.—"Power," Feb. 20, 1923.

NEW LEADIZING PROCESS FOR PIPING AND STEEL PLATES.

By C. C. Lance.*

Until recently no practical protective coating has been available for steel plates and piping subject to rapid corrosion. The maintenance cost of the unprotected pipe lines has been a serious item of expense to railroads and other large users of pipe. The galvanizing process affords no protection against acid fumes and the nature of the coating, which is only a surface deposit, makes it easily damaged, so that the coating will flake off when the pipe is struck or bent.

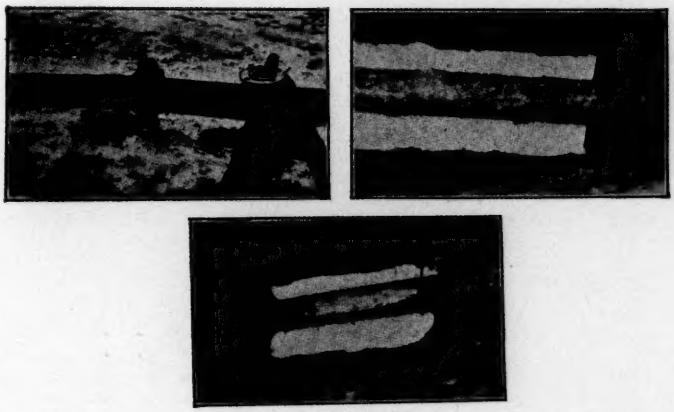
So much trouble was experienced by the Santa Fe Railway from the corrosion and pitting of locomotive boiler tubes, due to bad water conditions, that experiments were started to develop a remedy. Other sources of trouble were the piping and steelwork generally in railroad shops, roundhouses and other buildings where the gas from the locomotives penetrated. This gas, usually containing considerable sulphur, lodges under the roof and, combining with the moisture from steam vapor, forms a mixture containing sulphurous acid, which drips on piping and causes rapid corrosion. Steam lines properly insulated are not protected from injury, as the gas penetrates the smallest crevices. Outside of buildings around the yards, pipes buried in the ground or laid beside tracks are damaged by the chemical action of the cinders prevailing in these places. Some of the pipe lines corrode so rapidly as to require renewal within three years. These renewals are expensive and inconvenient owing to the 24-hour service usually existing. A wide variety of services are affected. For example, corrosion takes place in pipe lines for the conveyance of steam, compressed air and cold water, in boiler-washing and heating mains and returns, acetylene and oxygen lines, boiler-test lines and piping for special purposes as well as electric conduits.

It is not only in railway work, however, that rapid corrosion is prevalent. The average power plant is not free from troubles of this sort. The piping may be attacked externally or the chemicals found in some waters may corrode the interior surface. There are many places in the plant where protected steel piping at a cost on a par with wrought-iron piping would be highly desirable.

In the Santa Fe experiments lead seemed to be the best metal for the purpose on account of its resistance to corrosion and its long life. It was thought that, by combining lead with steel, or, rather, by coating steel with lead, a satisfactory protection might be evolved. This method was tried out with success, and an examination of some of the lead-coated pipes installed at Topeka in 1917 failed to show any depreciation, although in one instance the pipe examined had been buried in cinders for several years. Other piping was subjected to constant moisture in a drainage manhole with practically no sign of deterioration. The original lead coating remained intact, the only indication of service being a slight blackening of the surface.

The first step in the leadizing process is to remove the grease and oil from the material to be treated by immersion in a hot lye bath. After its removal it is washed with water and placed in a bath containing a 17 per cent solution of hot sulphuric acid. After the material is in the acid, about one pound of Chile saltpeter is added for each 40 square feet of surface to be cleaned. The saltpeter causes an eruptive action which thoroughly scours off rust and mill scale from the surface of the steel, leaving a clean surface.

* Engineer, National Boiler Washing Co., Chicago.



FIGS. 1 TO 3.—VIEWS OF LEADIZED PIPES BEFORE AND AFTER USE UNDER VARIOUS CONDITIONS.

Fig. 1—View of leadized pipe at Barrington shop. Fig. 2—Leadized iron pipe installed in 1917; notice moist earth sticking to it; pipe in excellent condition; no perceptible deterioration. Fig. 3—Main acetylene line installed in 1917; slimy portion has been scraped clean; no changes have been made in line; although bell and water seal of galvanized-iron acetylene generator near-by have been renewed three times during the same period on account of the rapid corrosion.

Following its removal from the sulphuric acid, the material is washed off with water and placed in a bath of hot acetate of lead. The acetic acid in the bath having greater affinity for steel than for lead, picks up steel and deposits lead, and at the end of twenty minutes, leaves a thin impenetrable homogeneous coating over the entire surface inside and out. As this coating insulates the steel, and longer immersion does not increase the thickness of the coating to a point suitable for general commercial uses, the material is removed and placed in a second bath containing acetate of lead and iron sulphate.

Use is then made of a regular plating generator, the material to be coated being connected to one pole and the lead anodes to the other. After an hour and a half in this bath, using a current of 1 ampère per square foot of surface, a coating of suitable thickness for all practical uses is obtained. In the same length of time, however, a coating of greater thickness might be obtained by increasing the current density. After removal from this final bath the coated material is washed off and painted, partly as it is a convenient time to do it and also to cover up the red streaks left on the surface of the coating by the iron sulphate in the bath. The paint adheres to the treated surface, as it has a natural affinity for the lead coating.

Examination of the treated surface shows an even coating of lead which is malleable and ductile and not subject to damage from bending or distortion, the lead, in fact, forming a component part of the material. As previously stated, the lead treatment protects against the corrosive action of sulphuric acid, but does not give entire protection against the action of alkaline solutions found in the water used by railroads in some parts of the country.

In the case of steel tanks for boiler-washing systems, the plates are first cut to size and punched, then leadized and bent to the proper radius before erecting. Lead-treated rivets driven cold are used, the formed heads being exposed to the water, but even the upset ends of the rivets show excellent penetration of lead.

Of the pipe lines used for the various services previously mentioned, some have been made up flanged and some screwed, but all have been fabricated at the Barrington shops, either cut to length and threaded before treating in the case of screwed lines, or made up to the required measurements for flanged lines. The leadizing treatment protects both flanged and screwed piping with equal satisfaction, making a particularly good coating for the threads, as it covers them as evenly as the smooth surface of the piping.

In ordinary piping the first signs of damage from corrosion are noticed at the threaded ends where the flanges or couplings are screwed on. This portion of the pipe is thinner, and as the outer surface has been removed when cutting the threads, the interior fibers are exposed and are more readily attacked by the action of the gases, so that the threaded portion may be eaten away while the remainder of the pipe may not be much affected. Galvanizing or painting steel pipe does not protect it against this action, so that it is often the case that such piping must be replaced as frequently as ordinary piping.

To overcome this trouble prior to the development of the leadizing process, a new type of flange connection for piping was designed, heavy drop-forged steel flanges without threads being used. This is shown in Fig. 4. The flange is shrunk on and finally welded to the pipe, and outlets are also welded to the pipe. In this way much of the work formerly done in the field is now transferred to the shops. Full measurements are taken on a job and straight piping is employed throughout any variation from a straight line being secured at the flange. The straight blank pipes are cut to the proper length and the extra-heavy drop-forged steel flanges are bored out

to the proper angle in a special adjustable jig. These flanges are heated and shrunk on the pipe and permanently welded in place. During the various operations gauges set to the proper angle are used to check the accuracy of the work, and before the finished job is turned out the faces of

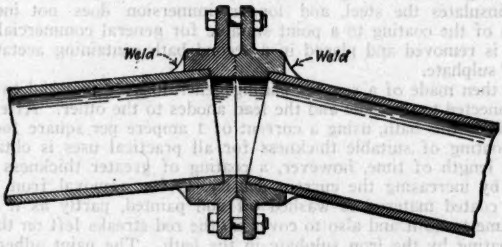


FIG. 4.—DETAIL OF SPECIAL WELDED DROP-FORGED FLANGE.

the flanges are ground off smooth in a grinding machine having an adjustable "V" block bed to give the smooth surface for the gaskets necessary to insure against leakage. As each installation is fabricated complete in the shop, the time required for installation is greatly reduced.—"Power," March 27, 1923.

DOUBLE ACTING DIESELS FOR NEW LINER.

SWEDISH-AMERICAN LINE PLACES ORDER FOR TRANSATLANTIC PASSENGER LINER OF 17,000 TONS GROSS AND 13,000 SHAFT H.P. ON TWIN SCREWS—FOUR-CYCLE ENGINES TO BE BUILT BY BURMEISTER & WAIN AND THE SHIP BY ARMSTRONG-WHITWORTH & CO.

In about eighteen months a large smokeless and probably funnelless passenger liner will cruise into New York harbor and drop anchor. She will be the 17,000 gross tons fast Transatlantic passenger vessel which the Swedish-American Line has just ordered from Sir Wm. Armstrong, Whitworth & Co., of Newcastle-on-Tyne, England, for delivery, September, 1924. The engines of this vessel are particularly noteworthy as they are of the double-acting four-cycle Diesel type and will be constructed by Burmeister & Wain of Copenhagen, who for some time past have been experimenting with a single cylinder 1,000 horsepower double-acting engine, as related in our last issue.

The new motor vessel is for service between Sweden and New York and will be approximately 560 feet long by 71 feet breadth and 29 feet draft. She will be the last word in naval architecture, so will be modern in all respects.

Accommodations for 1,660 passengers will be provided, represented by staterooms for 560 first and second-class, and quarters for 1,100 third-class passengers. The power to be installed is guaranteed to give a speed of 17 knots. It is anticipated the fuel-consumption will not exceed 50 tons per 24 hours compared with 130 to 150 tons for a steam turbine engined ship of the same power.

Announcement of the decision to use double-acting engines in this vessel has taken the marine world by storm and evoked much wondering comment. It has led also to a lot of foolish vaporing.

From an engineering standpoint there is absolutely nothing experimental in the construction of double-acting four-cycle engines, albeit the application to marine work of the experience gained with this class of machinery on land will represent a new achievement.

Double-acting Diesel engines of the four-cycle type have been used in stationary power plants for nearly 15 years past and have been giving eminent satisfaction. They have been built naturally only in the larger powers—from about 1,000 B.H.P. per crank and up. Over 10 years ago we inspected installations in Germany where such engines were operating, and found in one instance that a cotton-spinning mill was entirely dependent on one double-acting Diesel engine, with no other power unit in reserve and no possibility of getting electric power in an emergency. Evidence of that sort shows beyond cavil how reliable is the operation of such engines.

Long before the war started in Europe a number of the firms building marine Diesel engines over there had constructed single-crank double-acting four-cycle engines for test purposes. Polar, Nobel and Werkspoor each had one, and, if memory serves aright, Burmeister & Wain had already started the trials which have culminated in their obtaining the order for the machinery in the big vessel for the Swedish-American Line. Other firms went ahead with the construction of double-acting two-stroke engines in very large powers.

In no instance was any trouble encountered from the double-acting principle itself, nor is there any reason for expecting it. When a firm has a good single-acting Diesel engine, it can quite well go ahead with the double-acting type. Quite probably there will in each case be room for improvement of the mechanical details, because it is almost exceptional in any class of engineering to get every detail of a new design right at the first attempt.

So far as the new Burmeister & Wain engines are concerned, we are thoroughly convinced that there will be nothing experimental about them. We well remember the same firm building an eight-cylinder 400 B.H.P. engine to try out details of reversing before the *Selandia* engines were put through the shops. In the same way they have now been trying out a single cylinder double-acting engine of 1,000 B.H.P. to enable them to work out the details for the Swedish-American liner's engines. Burmeister & Wain never left any detail of design to chance.

The comment that has been broadcasted about special steels for these engines and higher pressures, etc., is all poppycock. The stresses in the double-acting type of engine are no higher than in the single-acting type. From an economic viewpoint the double-acting type is superior because it gives a better utilization of the metal used in the engine, and therefore permits the weight to be kept down.—"Motor ship," April, 1923.

THE LARGEST SULZER MARINE ENGINE.

THE FIRST STEPHEN-SULZER TYPE FOR A B.I.S.N. CARGO SHIP. THE USE OF TURBO BLOWERS FOR SCAVENGING.

The completion of the first Diesel engine which Messrs. Alexander Stephen and Sons have built is a matter of more than ordinary moment. This set, which is one of two for installation in the B.I.S.N. Co.'s 11,000-ton cargo ship *Dalgoma*, and which we were enabled to inspect last month through the courtesy of the builders, is, in the first place, the highest powered marine engine of the Sulzer type that has yet been constructed in this

country. The cylinder capacity is actually larger than that of the cylinders of the engines to be installed in the 20,000-ton liner now under construction at the Fairfield Yard. The diameter and stroke are 680 mm. and 1,100 mm. respectively against 690 mm. and 980 mm. of the Fairfield-Sulzer engine. The latter, however, runs at 135 R.P.M., while the Stephen engine will operate normally at 85 R.P.M., giving 1,600 B.H.P., the output at 100 R.P.M. being 2,000 B.H.P. Engines of exactly the same size and power have been built by Sulzers themselves at Winterthur for installation in the 11,000-ton cargo ship *Camranh* for the Chargeurs Réunis; this vessel will be completed towards the end of the year.

Another feature of the plant, which is seen in British engineering practice for the first time, lies in the employment of turbo scavenging. Before dealing with the main engines, of which two are to be installed in the ship, a few words may be given regarding this system. Instead of each engine driving its own piston scavenge pump, there are two rotary blowers, driven by electric motors, either machine being capable of supplying sufficient scavenging air for the two engines.

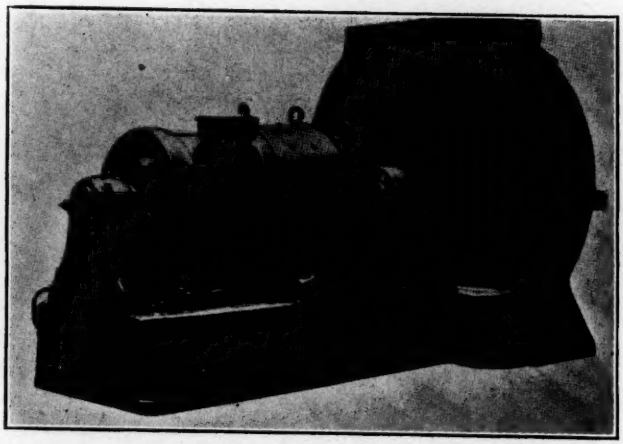
High speed of rotation is, of course, desirable for maximum efficiency of the blower and also to reduce the weight and cost. One of the two plants was supplied by Brown Boveri, while the other comprises a blower built by Reavells, coupled to an Oerlikon electric motor. The Reavell blower is a single-stage double-ended type, the impeller aspiring air on each side and delivering into a central volute-shaped casing arranged with a rectangular delivery connection on the top, the machine being therefore, balanced as regards axial thrust. It is capable of delivering 500 cubic meters of air per minute at a pressure of 1.15 atmospheres absolute, and runs at a speed of 3,000 R.P.M. It is direct coupled to a shunt-wound D.C. motor, developing 235 B.H.P. at this speed, and wound for 220 volts. The motor baseplate is rigidly attached to the compressor casing through a cast-iron distance-piece.

The two plants will be located in the thrust recess, and the scavenging air delivered through large pipes to the scavenging trunks on each of the engines. It is estimated that one blower, with its 150 horsepower motor, will be operating practically at full power the whole time when supplying air to the two engines in normal service.

The design of the new engine differs to some extent both from the Sulzer engines used on the *Conde de Churruca* and those built by Denny Bros., referred to on another page of this issue. Electrical auxiliaries are to be used throughout the ship, and the jacket-cooling water and piston-cooling water is to be supplied from separate electrically driven pumps in the engine-room. The battery of pumps usually arranged on the Sulzer engine at the forward end thus serves a different purpose from the normal. One of the horizontal pumps is utilized as a bilge pump, that opposite to it as a lubricating oil pump, and the vertical pump between them as a cross-head lubricating pump for supplying oil at a pressure of about 250 pounds per square inch to 300 pounds per square inch to the crossheads. This battery of pumps can be discerned at the left-hand side of the illustration, beyond the air compressor, which is also at the forward end driven off an extension of the crankshaft.

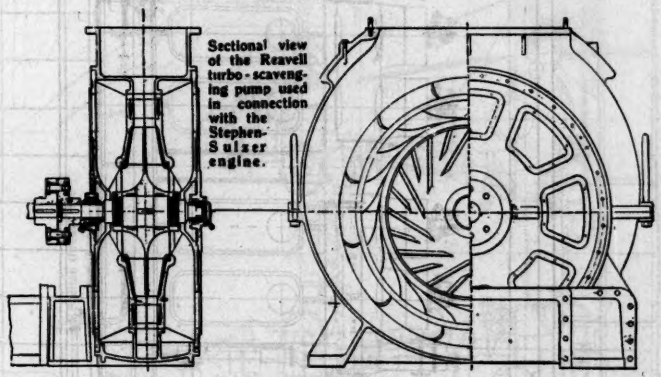
This compressor appears to be of more than sufficient size for the engine, having an output, we believe, of about 750 cubic feet per minute, while the requirements of the engine probably amount to about 400 cubic feet per minute. With most Sulzer engines this is usually over-designed, and in some ships, as, for instance, in the *Handicap*, the main purpose is to have an ample surplus for charging the starting-air bottles, since no auxiliary air compressor is carried.

THE ENGINE DEVELOPS 1,600 B.H.P. AT 85 R.P.M. AND 2,000 B.H.P. AT 100 R.P.M. (500 B.H.P. PER CYLINDER).



THE ELECTRICALLY DRIVEN REAVELL SCAVENGING PUMP.

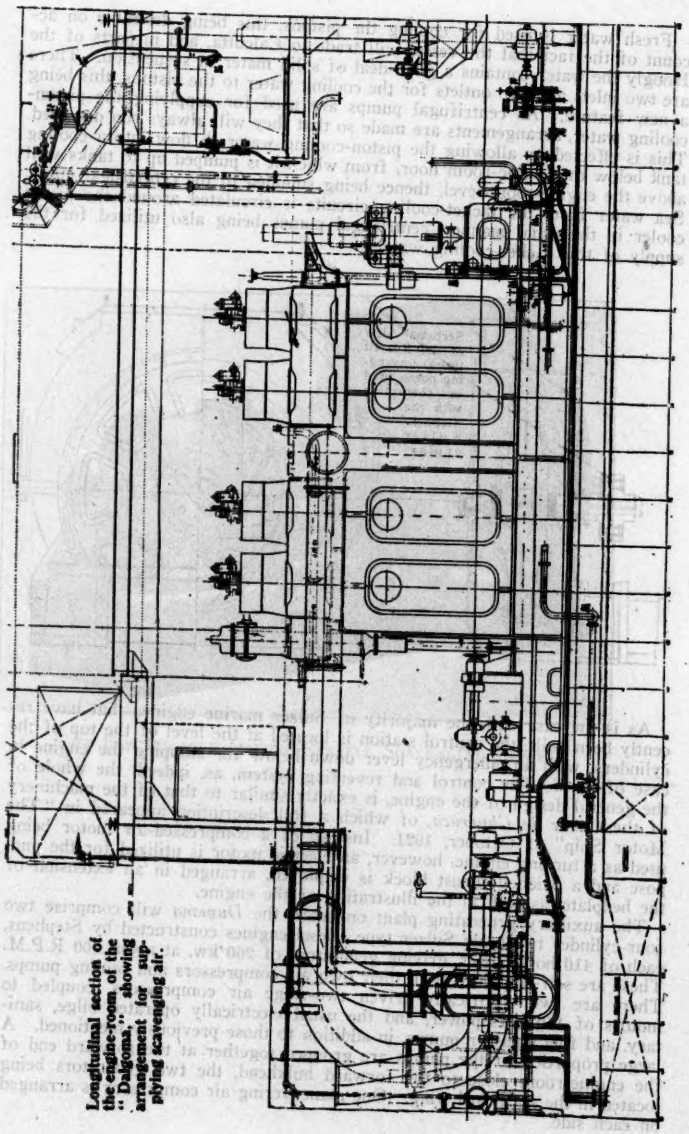
Fresh water is used for cooling the pistons, this being desirable on account of the fact that the vessel will trade to Calcutta, and in parts of the Hoogly the water contains a good deal of solid mater in suspension. There are two inlets and two outlets for the cooling water to the pistons, this being a new feature. As centrifugal pumps are used for supplying the piston-cooling water, arrangements are made so that they will always run drowned. This is effected by allowing the piston-cooling water to flow into a cooling tank below the engine-room floor, from which it is pumped up to tanks well above the engine-room level, thence being supplied to the centrifugal pump. Sea water from the jacket-cooling circuits is circulated around the piston cooler in the usual manner, centrifugal pumps being also utilized for the supply of this jacket-cooling water.



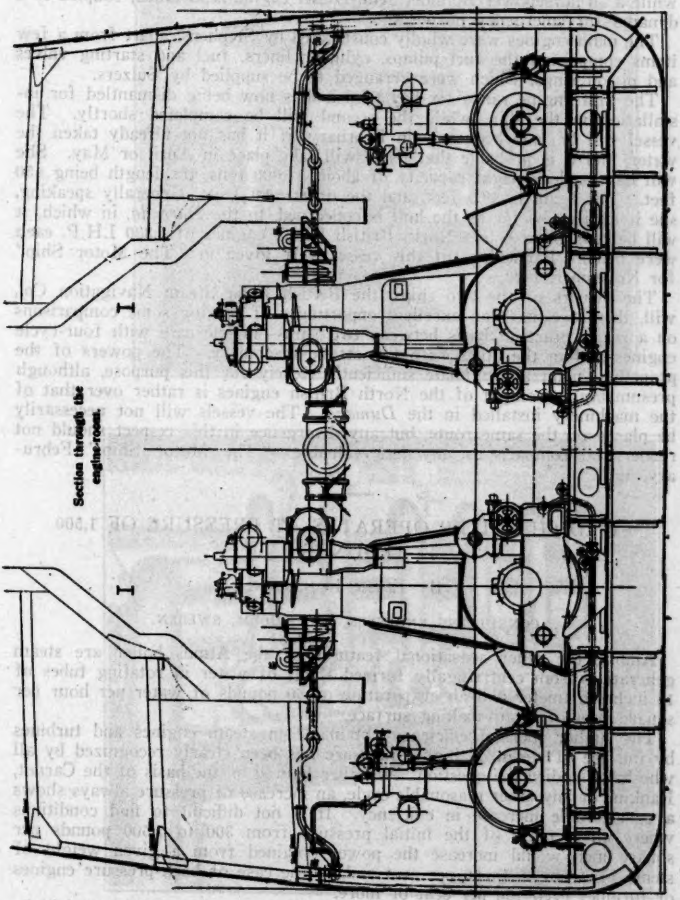
Sectional view of the Reavell turbo-scavenging pump used in connection with the Stephen-Sulzer engine.

As is the case with the majority of Sulzer marine engines that have recently been built, the control station is located at the level of the top of the cylinders, with an emergency lever down below for stopping the engine in case of need. The control and reversing system, as, indeed, the whole of the general design of the engine, is exactly similar to that of the machinery of the *Conde de Churruca*, of which a full description appeared in "The Motor Ship" of October, 1921. Instead of a compressed-air motor being used as a turning engine, however, an electric motor is utilized for the purpose and a Michell thrust block is employed, arranged in an extension of the bedplate, as seen in the illustration of the engine.

The auxiliary generating plant on board the *Dumana* will comprise two four-cylinder two-cycle Sulzer type Diesel engines constructed by Stephens, each of 410 horsepower, driving generators of 260 kw. at about 300 R.P.M. These are self-contained with their own air compressors and cooling pumps. There are two electrically driven two-stage air compressors, coupled to motors of 150 horsepower, and the usual electrically operated bilge, sanitary, and fuel transfer pumps, in addition to those previously mentioned. A large proportion of the pumps are grouped together at the forward end of the engine-room close to the forward bulkhead, the two generators being located in the starboard wing. One maneuvering air compressor is arranged on each side.



Longitudinal section of the engine-room of the "Dalgoma," showing arrangement for supplying scavenging air.



Section through the engine-room.

All the oil fuel will be carried in double bottoms, and arrangements are made to heat two of the tanks. The small donkey boiler for heating purposes is located approximately on the same level as the control platform, while a 30 horsepower Bolinder semi-Diesel engine is installed, coupled to a dynamo for emergency purposes.

The main engines were wholly constructed by Stephens, apart from a few items comprising the fuel pumps, cylinder liners, fuel and starting valves and piston rings, which were arranged to be supplied by Sulzers.

The first engine ran a six-day trial and is now being dismantled for installation in the ship, while the second will be completed shortly. The vessel will be launched early in February, if it has not already taken the water, and it is probable that trials will take place in April or May. She will have a deadweight capacity of about 11,000 tons, the length being 450 feet. The beam is 58.3 feet, and the draft 32.9 feet. Generally speaking, she is similar, so far as the hull is concerned, to the *Durenda*, in which, it will be remembered, two North British Diesel engines of 2,330 I.H.P. each were fitted. Particulars of this vessel were given in "The Motor Ship" for November, 1922.

The owners of the two ships, the British India Steam Navigation Co., will, therefore, have an excellent opportunity of making some comparisons on a really practical basis between two ships—in one case with four-cycle engines and in the other with two-stroke machinery. The powers of the propelling plant approximate sufficiently closely for this purpose, although presumably the output of the North British engines is rather over that of the machinery installed in the *Dumana*. The vessels will not necessarily be placed on the same route, but any divergence in this respect should not render the comparisons any less valuable.—"The Motor Ship," February, 1923.

SWEDISH BOILER OPERATES AT PRESSURE OF 1,500 POUNDS.

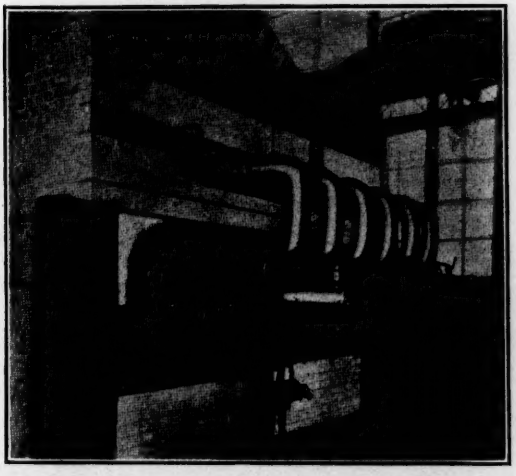
By Edvin Lundgren.

CONSULTING ENGINEER, STOCKHOLM, SWEDEN.

Among the other sensational features of the Atmos boiler are steam generation from centrifugally formed shells of water in rotating tubes of 12 inches diameter and an evaporation of 60 pounds of water per hour per square foot of steam-making surface.

The higher thermal efficiency obtainable in steam engines and turbines by the use of steam of higher pressure has been clearly recognized by all who have studied the question. Whether figured on the basis of the Carnot, Rankine or any other reasonable cycle, an increase of pressure always shows a considerable increase in efficiency. It is not difficult to find conditions where an increase of the initial pressure from 300 to 1,500 pounds per square inch would increase the power obtained from a given weight of steam by from 20 to 60 per cent and, in the case of back-pressure engines or turbines even 100 per cent or more.

Fig. 2 shows how large are the savings made possible by the use of high-pressure steam, particularly in the case of back-pressure prime movers. In this diagram, N_1, N_2 and N_3 denote the brake horsepower that may be obtained from 1,000 pounds of steam per hour during its expansion from an initial pressure of 1,500, 300 and 220 pounds (gauge) respectively down to various back pressures. In all three cases it has been assumed that the steam in its initial state had the same heat content of 1,370 B.t.u. per pound.



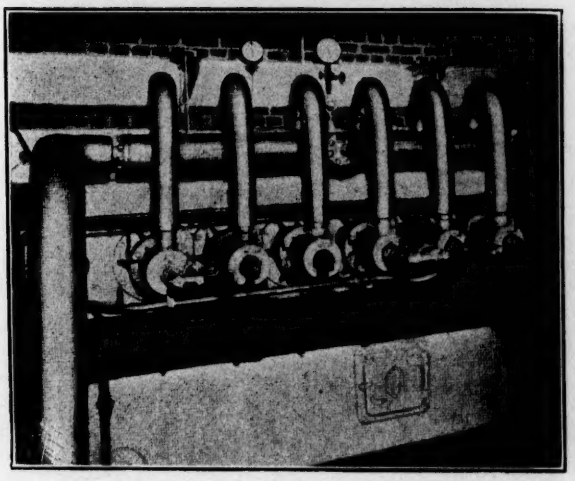


FIG. 1.—REAR VIEW (STEAM END) OF ATMOS BOILER INSTALLATION AT CARNEGIE SUGAR REFINING WORKS, GOTENBURG, SWEDEN.

The picture clearly shows the steam connections to the six rotating tubes. This boiler has been in continuous daily service for more than a year, carrying a steam pressure of 900 pounds per square inch. At the top of the page is a view of the front of the same boiler. This shows the motor which rotates the tubes at 330 R.P.M. by means of gears attached to the ends where the feed water enters.

The curves t_1, t_2 and t_3 indicate the respective temperatures of the exhaust steam leaving the prime mover, while t_s is the temperature of saturated steam at the various exhaust pressures.

On account of the lower initial superheat and the greater work done in expansion, it is easily understood that the exhaust steam from a turbine using the high-pressure steam is less superheated (or, if saturated, has a lower quality) than that of a turbine with a lower initial pressure. However, if the exhaust steam is to be used for industrial heating purposes, this may frequently be an advantage.

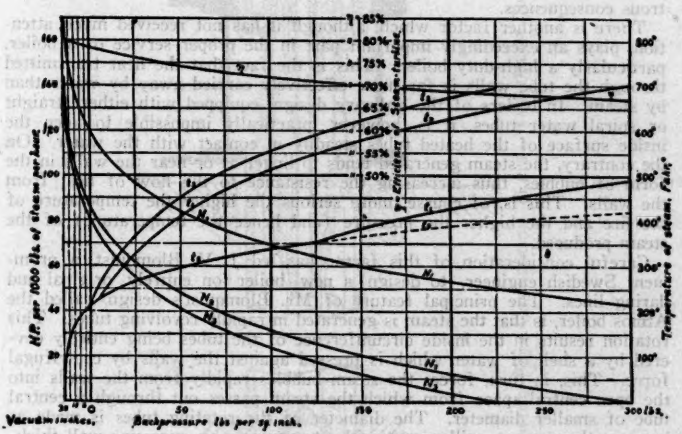


FIG. 2.—DIAGRAM SHOWING EFFECT OF INITIAL AND FINAL PRESSURES ON POWER DEVELOPED.

The curves N_1, N_2 and N_3 show the power developed expanding 1,000 pounds (weight) of steam to various back pressures, where the initial pressure is 1,500-pound gauge for N_1, 800 pounds for N_2 and 220 pounds for N_3. In each case the initial superheat is such that the total heat of the steam is 1,370 B.t.u. per pound. The curves t_1, t_2 and t_3 give the exhaust temperatures corresponding to N_1, N_2 and N_3. Both sets of curves are figured on the basis of the Rankine efficiencies given by the curve marked n.

That, wherever possible, steam in its higher pressure ranges should be used for generating power in back-pressure engines and turbines, and the exhaust steam from the prime movers used for industrial heating purposes, is more and more realized by steam engineers. A glance at Fig. 2 shows the importance of high initial pressure when back-pressure turbines are required to deliver much power.

In recognition of its advantages the modern trend has been steadily toward the application of higher pressures. This has undoubtedly helped to make the water-tube boiler predominant in the larger modern steam plants. Certain difficulties, however, as well as safety considerations, have prevented the general adoption of pressures higher than 350 pounds, at least in stationary practice. The recent use of 400 pounds in the new central station of the Public Service Company of Northern Illinois at Waukegan and 550 pounds at the Philadelphia plant of the Ohio Power Co. are exceptions that indicate the trend of modern power-plant design.

In smaller boiler sizes attempts have been made, by De Laval, Herreshoff, Serpollet, White and others, to produce steam at higher pressures. In most of these cases spiral tubes of a small diameter were used, through which the water was forced. The small diameter of the tubes allowed small thicknesses of the tube walls—a favorable condition for heat transmission. Moreover, the danger in case of an explosion was greatly reduced on account of the small water content of the tube. On the other hand, it is evident that the removal of sediment and harder incrustations from the inside walls of the small tubes is a difficult if not impossible task. Such scale incrustations might easily cause a local overheating of the tubes with disastrous consequences.

There is another factor which, although it has not received much attention, plays an exceedingly important part in the proper service of a boiler, particularly a high-duty boiler. This is the fact that the heat transmitted through the tube walls is far more effectively carried away by water than by steam. In boilers of the ordinary design, equipped with either straight or spiral water tubes, it is, however, practically impossible to keep the inside surface of the heated tubes steadily in contact with the water. On the contrary, the steam generated tends to linger at or near the walls in the form of bubbles, thus increasing the resistance to the flow of heat from the walls. This is, of course, more serious the higher the temperature of the fire and the higher the pressure (and hence the temperature) of the steam produced.

Careful consideration of this factor has led J. V. Blomquist, a prominent Swedish engineer, to design a new boiler on entirely original and daring lines. The principal feature of Mr. Blomquist's design, called the Atmos boiler, is that the steam is generated in rapidly revolving tubes. This rotation results in the inside circumference of the tubes being entirely covered by a shell of water which is pressed against the walls by centrifugal force. This, in turn, forces the steam bubbles rapidly from the walls into the open central space from which the steam passes out through a central tube of smaller diameter. The diameter of the rotating tubes is made as large as the pressure will permit without requiring an excessive wall thickness.

Moreover, the construction is such that the rotating tubes, or the "rotors," as the inventor calls them, may expand freely without strain. In ordinary boilers it is impossible to avoid those stresses, and it is even impossible to determine them accurately.

The effective cooling of the rotor walls and the freedom from expansion strains make possible rates of evaporation as high as 60 and sometimes even 100 pounds of water per hour per square foot of heating surface. This makes for an exceedingly compact construction.

How the solution of the underlying problem is carried out in practice and a reliable boiler for extremely high working pressures is obtained is best explained by referring to the accompanying drawings and photographs, from which it will be seen that the design embodies a number of rather ingenious features.

The headpiece and Fig. 1 show photographs of an actual installation of which more will be said later.

Fig. 4 shows the general arrangement of a double boiler in plan, elevations and various sections. Each of the two boilers is calculated to generate about 16,500 pounds of steam per hour at 1,500 pounds pressure and 100 degrees superheat (total temperature 700 degrees) and is provided with eight rotors which have an outside diameter of 1 foot and ¾-inch-thick walls. The length of the parts of the rotors directly exposed to the heat of the fire is about 11 feet 2 inches. The total heating surface of each

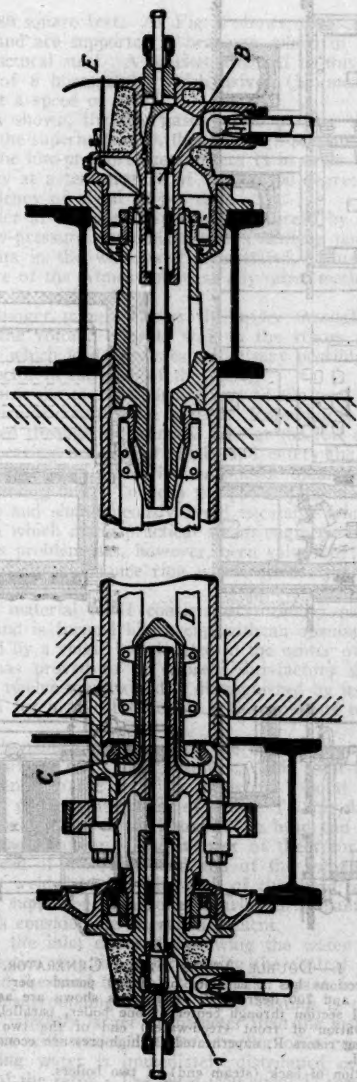


FIG. 3.—SECTION THROUGH A ROTOR, SHOWING FEED CONNECTION (LEFT) AND STEAM CONNECTION (RIGHT).

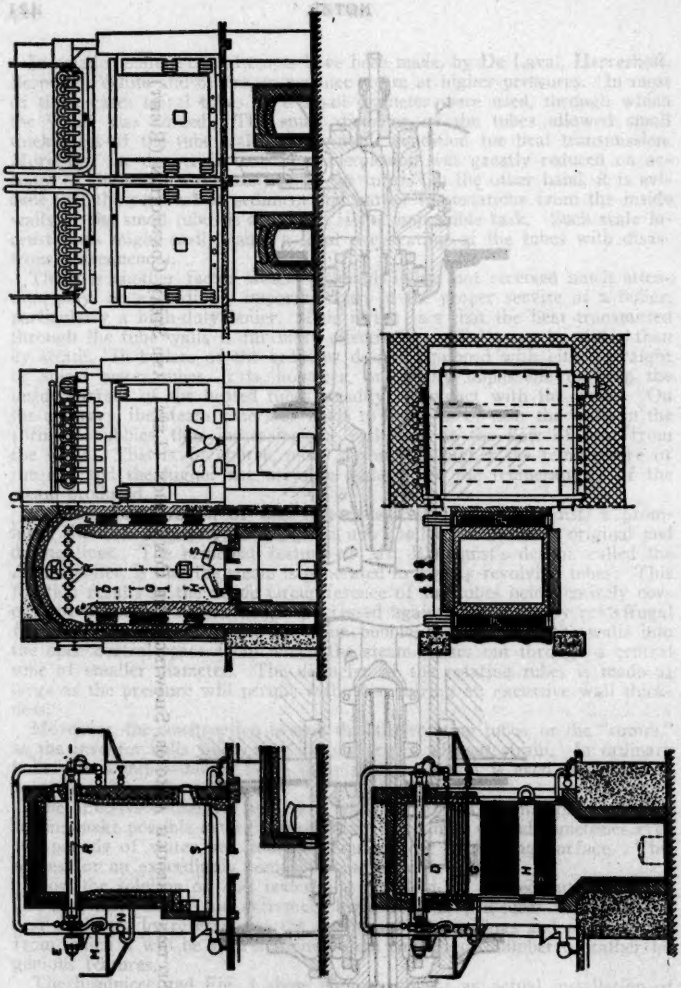


FIG. 4.—DOUBLE ATMOS STEAM GENERATOR.

Each of the two sections has a capacity of 16,500 pounds per hour of steam at 1,500 pounds pressure and 700 degrees F. The views shown are as follows:

Upper left: Vertical section through center of one boiler, parallel with rotors R.

Upper center: Elevation of front (feed-water) end of the two boilers with one boiler in section showing rotors R, superheated D, high-pressure economizer G and low-pressure economizer H.

Upper right: Elevation of back (steam end) of two boilers.

Lower left: Vertical cross-section through F (see figure in upper center) showing rotors, superheated D, high-pressure economizer G and low-pressure economizer H.

Lower center: Top view of two boilers, with one boiler in cross-section just above the superheaters.

boiler is thus 280 square feet. As Fig. 3 shows, the rotors extend through the brickwork and are supported in bearings, which in turn are carried by a frame of structural steel. A bracket attached to this frame supports an electric motor of 8 horsepower which drives (by means of spur gears) all the rotors at a speed of 330 R.P.M.

In the design shown, the flue gases, after passing the rotors, are first utilized to heat the superheaters D, then the high-pressure water economizers G and, finally, the low-pressure economizers H in order thereafter to escape into the chimney at a temperature of 350 to 400 degrees F., corresponding to a boiler efficiency of about 80 per cent.

The feed water at 150 pounds pressure is forced by a centrifugal pump through the low-pressure economizer into a settling tank where the scale-forming elements in the water are precipitated. This is, of course, not strictly a feature of the Atmos boiler, as any other method of softening the feed water may be employed.

A multiple-plunger pump delivers the water through the high-pressure economizer to the rotors. On its way to the rotors the water passes a steam separator which leads any steam that may possibly have been formed to a steam collector in the rear of the boiler.

The rotors form the most important part of the new invention, and their design is therefore of considerable interest. Referring to Fig. 3, the hot feed water is admitted through the vertical pipe at the left end to the cover of the stationary bearing housing or shield and enters the rotating part at A in the central pipe shown, which has an outside diameter of 1½ inches and revolves in a stuffing box. To keep the packing tight against the pressure of 1,500 pounds and simultaneously avoid excessive friction, is of course a serious problem which many practical steam engineers will consider apprehensively. This problem has, however, been solved by admitting oil under pressure to the central distance ring which divides the packing longitudinally into two parts.

The packing material itself consists of ordinary packing braids. The stuffing-box gland is formed like a cap with an opening on the lower side and is tightened by a single bolt located in the center of the bearing shield. This packing has proved to be entirely satisfactory during a continuous service of over twelve months and is distinguished by its low friction. The oil consumption for a 12-rotor double boiler amounts to about one pint in ten hours.

The bearing housing on this end of the rotor contains a ball bearing in which the neck of the rotor head revolves. This rotor head is a steel casting and is fastened to the rotor by means of a solid flange on the head, which is kept in place by a number of bolts. To this flange is attached the spur-gear ring which drives the rotor. The head can be readily removed for inspection and cleaning of the interior of the rotor.

The outlet head at the right-hand end of the rotor is screwed into the rotor and then welded to it. The neck of this head, and with it this end of the rotor, is supported by a roller bearing in a manner that allows free expansion and a considerable lateral movement.

Returning to the inlet end and following the water on its passage into the rotor, we notice that the water leaving the central pipe flows back (to the left) toward the head through the annular space formed by the hub of the head and a surrounding casting. This casting is provided at one end with a flange C, slightly larger in diameter than the inside of the rotor and fitted on its face with vanes like the impeller of a centrifugal pump. By this the incoming water is immediately distributed over the whole inner circumference of the rotor. The sheet-iron strips D serve to start the water rotating quickly when the rotor is started up.

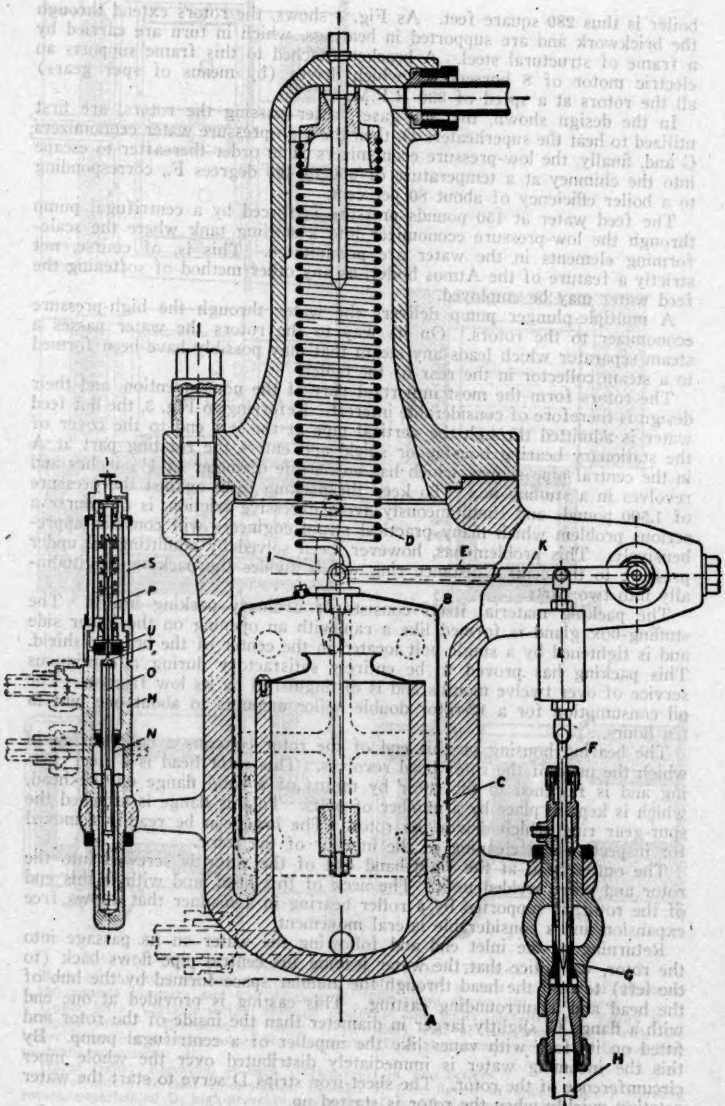


FIG. 5.—FEED-WATER REGULATOR AND GAUGE GLASS.

As mentioned before, the steam generated is collected in the interior of the rotor and escapes through the central-steam-outlet pipe (at the right in Fig. 3) which revolves in a stuffing box of the same construction as that on the water-admission tube.

The chief purpose in designing the flange C like the impeller of a centrifugal pump was to furnish means for measuring the thickness of the water shell and keeping it in correct proportion to the required rate of evaporation. On account of the centrifugal action the pressure in the water inlet is less than that prevailing in the steam outlet, the difference depending on the thickness of the water. Small pipes transmit the water-inlet and steam-outlet pressures to the water regulator shown in Fig. 5. The water pressure is transmitted to the inside of a bell B which is suspended in a closed casing A by means of a spiral spring D, the lower edge of the bell being submerged in a mercury seal C. The whole regulator, inside as well as outside of the bell, aside from the mercury seal, is filled with cold water. The steam pressure acts on the outside of the shell. The difference between the two pressures acts through the linkage EKL to move the needle valve FG controlling a by-pass from the delivery pipe to the suction line of the high-pressure feed pump.

When the load or the rate of evaporation increases, the thickness of the water body in the rotor decreases, which results in a corresponding change in the differential pressure. This acts on the bell and levers to cut down the bypass flow, thus increasing the flow of feed water to the boiler. The regulator continues to act in this manner until perfect equilibrium is attained. To each load or evaporation there corresponds a certain thickness of the water layer, varying between 1⅛ inches and 2 inches. This regulator is stable in its action and the resistance of the needle valve is small. The change of the net force acting on the bell in the two extreme positions amounts to 45 pounds. As all rotors are connected, one regulator is sufficient for the whole boiler.

It is, of course, perfectly feasible to accomplish the water control in some other way; for instance, by influencing the speed of the motor that drives the feed pump.

As explained before, the thickness of the water body in the rotor corresponds to a certain position of the bell. This position may serve as a water gauge, as it is easily indicated on the outside of the regulator by an extension of the lever K. For the sake of safety and convenience, however, the water is gauged by an independent water gauge attached to the regulator housing as shown at the left in Fig. 5. This water gauge is operated by the same differential pressure that controls the regulator. The narrow cylindrical casing of the gauge is divided into two compartments. The lower one, partly filled with mercury, is connected to the water pressure, while the upper one is connected to the steam. The two compartments are connected by the stationary central tube N, the lower end of which is immersed in the mercury. The varying level of this mercury indicates the thickness of the water body in the rotor and is read at S from the top of a steel float O.

The water gauge, like the regulator, is filled with cold water so that the risk of bursting the glass P is slight. Should such a thing happen, however, the steel rod O will be thrown upward and its tapered seat T will close the opening leading to the exterior. If, by any accident, a rotor should fail or burst, the ball check valves (shown at the ends of the feed and steam connections in Fig 3) automatically shut off the damaged rotor from those under pressure.

Another safety device is the elongation indicator E shown in Fig. 3 at the right. One end of this indicator lever is in contact with the moving rotor end, while its fulcrum stays stationary with the bearing housing. If

the rotor expands, the outer end of the lever E will indicate the exact amount of elongation. If scale should be formed and cover the inner surface of the rotor, it would increase the resistance to heat flow so that the temperature of the rotor walls would rise abnormally. This, in turn, would cause an undue elongation of the rotor which would immediately be shown by the indicator, thus warning the boiler attendants in due time that a cleaning of the rotor would soon be necessary.

All the devices and constructions described have passed the first experimental stage and have stood the test of actual service. The headpiece and Fig. 1 show views of the first experimental installation located at the Carnegie Sugar Refining Works, at Gotenburg, Sweden. Here the steam is generated at a pressure of 900 pounds per square inch. The boiler has been in continuous daily service under rather unfavorable conditions ever since December, 1921, and has given entirely satisfactory results.—"Power," February 13, 1923.

A PROPOSED AIRCRAFT CARRYING MAIL STEAMER.*

By Sir Eustace D'Eyncourt and John H. Narbeth.

In describing "Naval Construction During the War," before this Institution in 1919, some reference was made to the production of aircraft carriers for naval purposes, and it was stated, "There is no question that aircraft carrying will gradually become more important for the Fleet." The few years which have since elapsed have fully confirmed that opinion, and the value of the aircraft carrier for naval purposes is indicated by the large sum of money voted in the current Navy Estimates for that class of work.

The experience gained in the naval aircraft carriers indicates that flying on to, and off from, ships at sea is rapidly becoming thoroughly practicable. Some six hundred successful flights on to H.M. ships *Argus* and *Eagle* have already been made, under varying conditions of weather, at sea.

It may well be that we are now advancing to a position in which the aircraft carrier will become relatively quite as important a ship for the mercantile marine, and that with the growth of opinion and technique, shipowners will presently be prepared to face the bills which will arise from the adoption of a new type of vessel of this class, for commercial purposes.

At the Air Conference held at the Guildhall in the autumn of 1920 a suggestion was made by one of the principal speakers to consider the practicability of carrying aircraft on mail steamers. At that time some experience had been gained in fitting up mail steamers, *e.g.*, the Australian Government steamer *Nairana* had been fitted out as a seaplane carrier. Stowage was provided for seven seaplanes, and derricks were provided for hoisting them in and out. This, however, was a very cumbersome method and totally different in conception from the aircraft carrier as specially developed for naval purposes.

The carrying of aircraft in mail steamers had also been approached at that time from quite a different point of view, viz., as a supplementary naval service, and the Director of Naval Construction was able to state that an outline design had been prepared for an aircraft-carrying mail steamer, which could accommodate about twenty aeroplanes of moderate size, in peace time, and a larger number, if required, in case of war.

The question has now been further investigated, and drawings are attached to this paper showing what is hoped may be the basis for a practical discussion on the possibility of an aircraft carrier regarded, not as a supplement to naval services, but as a special development of the mercantile

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* Institution of Naval Architects, March 21st, 1923.

marine; and, by the methods described later, it will be seen that, in the proposed vessel, which is of quite moderate dimensions as compared with the leading Atlantic liners, besides providing all the facilities required to complete the vessel as a commercial, mobile aerodrome, there will remain ample deck spaces for accommodation of all kinds and particularly an enormous open upper deck available for recreation, besides large protected areas on the next deck beneath it. The cabin accommodation provided is about 80 per cent of that of *Mauretania* for first and second-class passengers, and 40 per cent for third-class passengers, although the vessel is much smaller, and large spaces are set apart for aircraft. It is hoped that the proposals will be found worthy of the careful attention, not only of the members of this Institution, but of the ship-owning interests of this country.

The combination of nautical and aeronautical craft for a common purpose is by no means an unreasonable proposition; their relations deserve careful study, as will now be shown.

The idea of the aeroplane serving as handmaid of the merchant ship is already in vogue to a limited extent. In case of need, passengers now occasionally leave London or Paris by aeroplane some time after the hour when the mail trains leave. The aeroplanes get to an aerodrome at the port in time to enable the passenger to alight and then embark on the liner with which the mail train connects, thus giving a very notable advantage in time saved to the traveller. The advantage will obviously be increased if the aeroplane, instead of alighting at an aerodrome and requiring the passenger then to proceed by some other conveyance to the ship, possibly by motor car and also by ferry steamer, could also fly right on to the ship herself while she is proceeding on her voyage at sea. Should this be practicable, flying on and off ships at sea opens up a great vista of future developments for combined air and steamer services.

Before mentioning the possibilities of some of these future developments, it is perhaps well to consider the practicability of building mail steamers in such a way as to offer the facilities necessary for accommodation of aircraft and at the same time for the accommodation of passengers, mails, and cargo, so that the whole combination might be run as a commercial concern. This proposal is put forward entirely as a commercial proposition and not as a subsidiary naval vessel. This point is mentioned at this stage because the proposal embodies a machinery arrangement based on naval designs, but that is the only part of the vessel which can be regarded as in any sense related to the Navy. It is fully recognized that the nearer the vessel can approach to ordinary commercial purposes the better, and it is hoped that in the discussion which will take place on the paper that this aspect will be thoroughly discussed and criticized with the greatest freedom.

The vessel proposed as a basis for this discussion is 600 feet long between perpendiculars with a beam at the water line of 80 feet, a mean draught of 28 feet, and a displacement of about 24,000 tons when leaving port fully laden with about 2500 tons of fuel and 1500 tons of cargo. Water-tube boilers are provided for, capable of generating steam for about 50,000 horsepower; together with geared turbines and twin screws absorbing 40,000 to 45,000 horsepower, and giving a speed of 24 knots on trial at a mean draught of 26 feet. The arrangements proposed, with steam-driven propelling machinery, involve the great drawbacks incidental to large funnels and uptakes, but the general design of the vessel would remain if it were possible to fit her with internal combustion engines of the requisite capacity. The suggestion to build aircraft carriers with Diesel engines for propelling machinery has often been made, but up to the present date no engines of the required capacity have been produced. The largest yet ordered for liners are only about half the power required for this vessel.

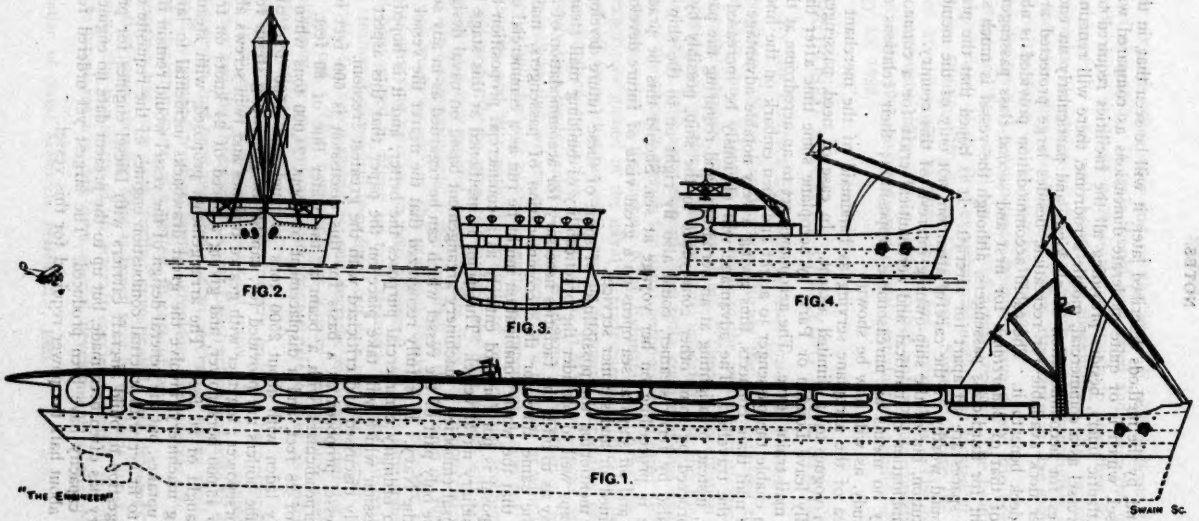


FIG.2.

FIG.3.

FIG.4.

FIG.1.

"THE ENGINEER"

SWAIN Sc.

DECK ACCOMMODATION.

The flight, or "A" deck, is approximately rectangular, being 565 feet long and 100 feet wide throughout, and it is entirely free from all obstructions. The deck finishes off practically square at a position 100 feet from the bow. The tapering end of the deck forward, which has been omitted, would appear to serve but little purpose, whilst its omission enables very satisfactory navigating and cable arrangements to be made. A fore mast is proposed, to carry the navigation lights, and also to carry long derricks for hoisting of cargo, aircraft, stores, baggage, &c., to the flight deck, and thence to the holds. The mast arrangement may appear at first sight open to objection as being in the way of aircraft, but this view is not sound, for machines rise in a very few yards, and after a very short time are well in the air and can be maneuvered to either side of mast and shrouds, without any trouble; while for landing there is a flight deck 565 feet long, and in case of failure to alight the machine would get away clear of the vessel long before reaching the mast position over the bow.

An alternative scheme has been prepared—Fig. 4—showing what would be involved if the fixed mast is not adopted, substituting therefor close stowing cranes for handling cargo, aircraft, &c., and a telescopic mast for carrying the steaming and navigating lights.

The second, or "B" deck, is more or less open fore and aft to permit of some flow of air along the underside of the flight deck, assisting to give a uniform flow of air above the flight deck. At the forward end is situated the navigating bridge, which is about 35 feet above water and well protected from the weather. On each wing of the bridge is a look-out tower for sighting aft, working the docking telegraphs, &c. Abaft the officers' quarters is a block of lifeboats, twelve in number, which can be launched from either side, and abaft this is the lift serving the forward hangar, while the space around the lift opening forms an aircraft assembling space. Flaps are arranged abreast the lift, in the flight deck. When these are lowered, they provide wind protection for the purpose of rigging the machine, and the openings enable the spread machine to go to the flight deck ready for flight. The lift machinery is also fitted on this deck. Abaft this lift is the second block of twelve lifeboats arranged as described above, and abaft these lifeboats is the second lift and assembling space arranged as before. Abaft the second lift an emergency oil-driven dynamo is fitted, and behind the dynamo the smoke ducts rise from the boilers and fill in the amidship space aft.

The third, or "C," deck, which runs right to the bow, is the first cabin deck.

The design provides for bulges with a view of obtaining some protection by them against results of damage by icebergs, collision, or other misadventure in peace time, and as a protection against the effects of mines, torpedoes, &c., in war time. They are arranged to carry oil when required for specially long voyages, thus supplementing the oil carried in the double bottoms and wing compartments to practically any desired extent. By making the bulges an integral part of the ship an efficient "form" can be designed in the first place, so that they do not involve loss of speed, or, stated in another way, they do not increase the horsepower required for a fixed speed.

An incidental, but very great, advantage arises from the adoption of the bulge. It enables a very broad vessel to be built without involving excessive initial stability, or undue resistance, and so permits large deck areas to be provided at an economical cost for production, far less, indeed, than if the vessel were of the usual proportions, and had to be increased in length to give the same deck areas. It should be remembered, however, that a very broad flight deck is essential.

Although the addition of the bulge will make the ship very broad beneath the water, and will provide the means for carrying very broad decks above the water, the breadth at the water line itself is not so great, and this will avoid the necessity of giving the ship too much initial stability. The initial stability, as measured by the metacentric height with the ship at rest in the various conditions of lading, will be kept as small as possible in order that the ship may not be a quick roller. The formation of the bulges, with the addition of bilge keels, will provide considerable resistance to rolling. This resistance, taken in conjunction with the small metacentric height, will make the ship very steady at sea, but it is proposed, in addition, should it be found desirable, to make use of the most efficient stabilizing devices which may be obtainable when the ship is built.

A steady ship is, of course, of the greatest attraction to passengers, and is most desirable also from the aviation point of view.

CAPACITY FOR AIRCRAFT.

The capacity for carrying aircraft depends upon the type and construction of the aircraft, as well as their extreme dimensions. It would be necessary to modify existing commercial types by fitting the wings to fold back, or the tails to hinge, in order to be able to adopt lifts of a reasonable size and to stow a good number of machines of sufficient carrying capacity.

In the absence of such types an estimate of capacity for stowing naval types of aircraft has been made with the results shown in Table II.

TABLE II.

Machine.	Forward hangar.	After hangar.	Total.
Torpedo carrier (Sopwith Cuckoo).....	10	8	18
Parnall Panther	11	10	21
Nieuport Night Hawk.....	12	9	21

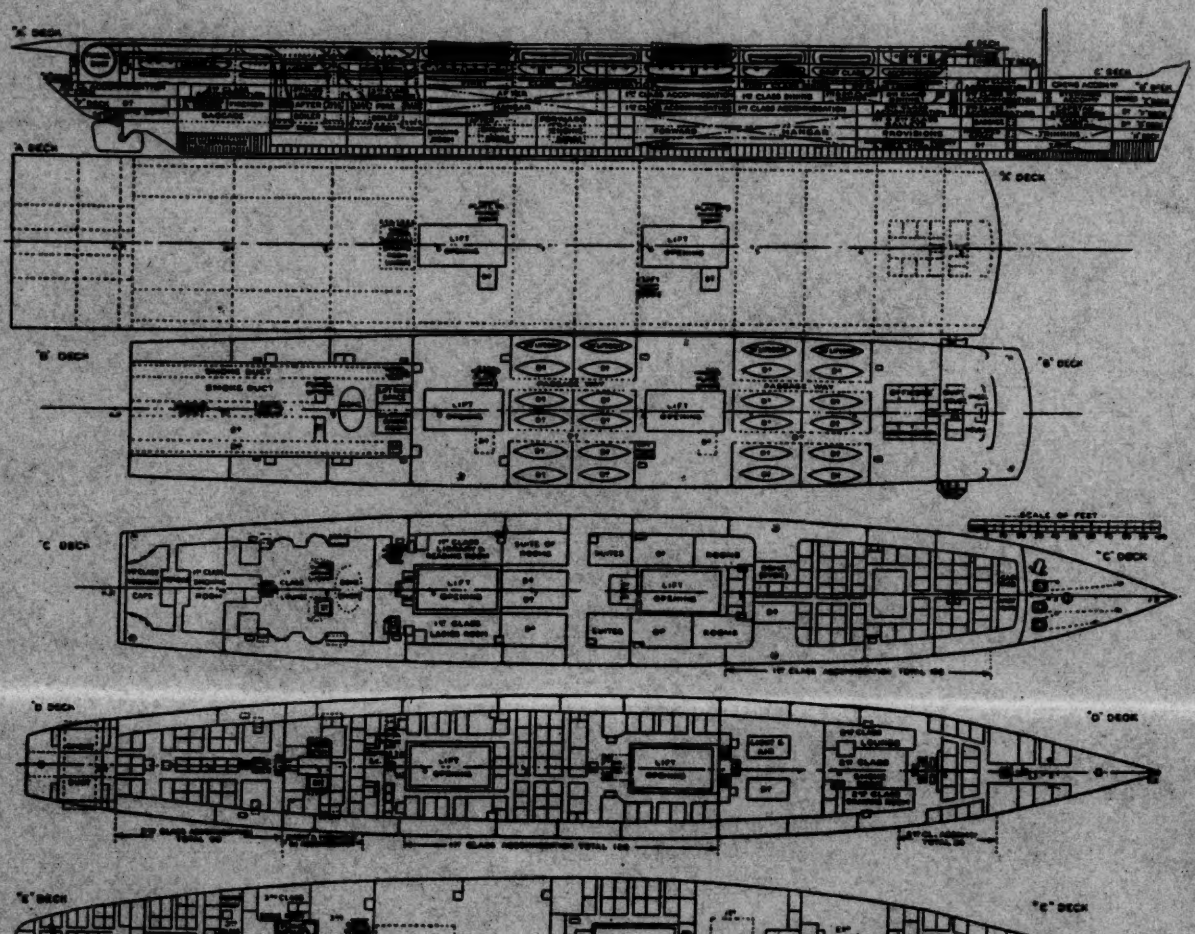
In making these estimates the machines are assumed to be folded, or stowed closely, but leaving the lifts empty and a clear space for one machine near each lift. The stowages are alternative, *i.e.*, in each case the hangar is completely occupied by one type of machine only, as shown in Column 1. When the demand is small only one hangar need be used, giving room for eight to twelve machines, while if both hangars are used, from eighteen to twenty-two machines of the types named can be taken.

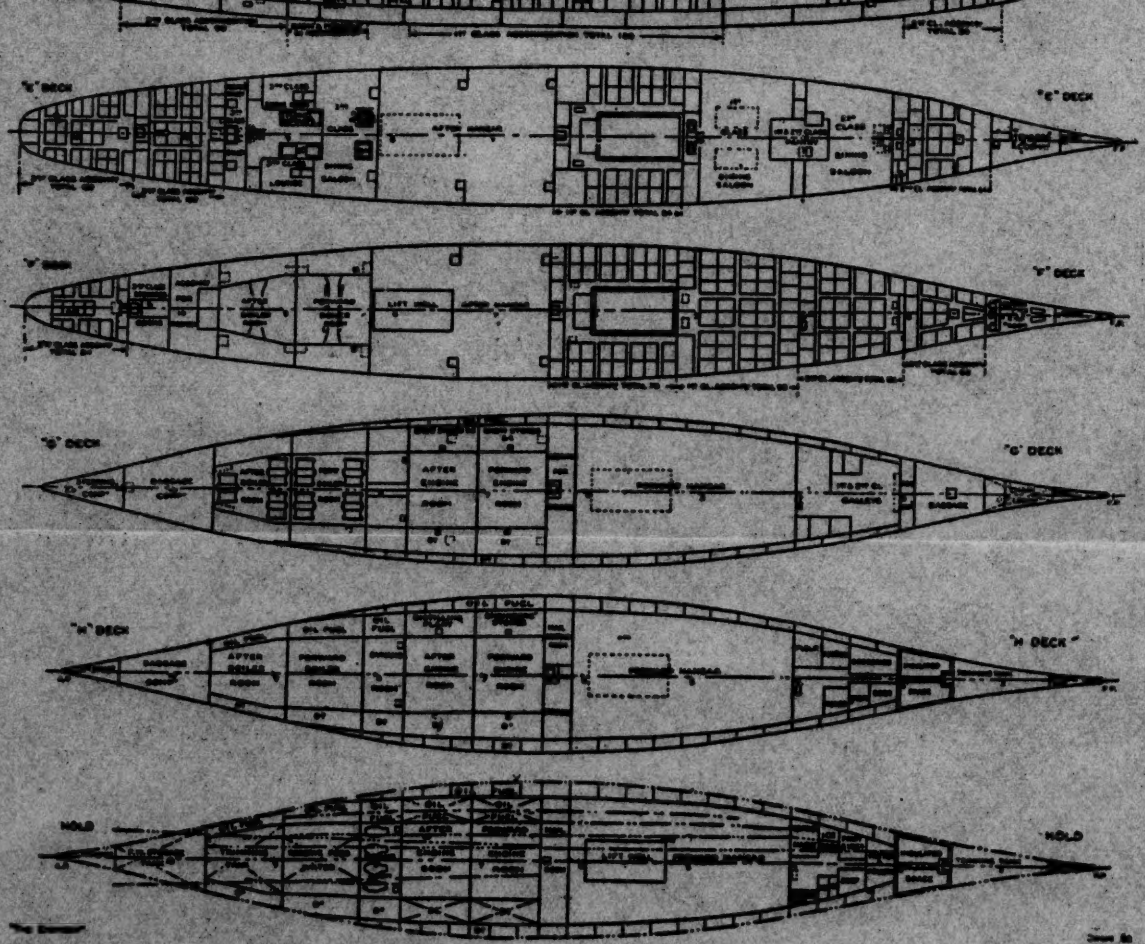
The outside dimensions of the machines taken compare with commercial machines as shown in Table I.

It would probably be found in the course of a short time that "joy rides" from the ship would be very profitable, and small two-seater machines could be provided for such a purpose. By this means on many routes passengers could view towns and country of great interest from the air, without going ashore, or stopping the vessel. On a yachting expedition to Norwegian fiords such a ship with a squad of machines for this purpose would prove a great attraction. In addition, relatively slow machines of various types might be found to be very economical if the route is such that only short trips away from the ship with passengers, mails, or baggage are necessary.

ADVANTAGES OF PROPOSED TYPE OF VESSEL.

Should these proposals be found acceptable and commercially practicable, they will not only expedite the present mail and passenger services, but will offer facilities for rapid communication with ports or inland localities which have hitherto been very badly served by offshoots from the regular passenger and mail services. For instance:—


'A' DECK
'B' DECK
'C' DECK
'D' DECK
'E' DECK
'F' DECK
'G' DECK
SCALE OF FEET



PROPOSED AIRCRAFT CARRYING MAIL STEAMER.

TABLE 1.

Type.	Dimensions spread.		Dimensions folded.		Weight, light.	Load, ex fuel.	Maximum speed.	Endurance (approx.).
	Length.	Span.	Length.	Span.				
	ft. in.	ft. in.	ft. in.	ft. in.	lb.	lb.	m.p.h.	hours.
MILITARY MACHINES—								
Torpedo carrier (Sopwith Cuckoo) ..	28 4	46 8	28 4	16 9	2160	1280	97	3½
Parnall Panther	25 0	29 5	14 6	29 5	1328	726	108	4¼
Nieuport Night Hawk	19 6	28 3	10 6	28 3	1552	398	139	1¼
Flycatcher	23 1	29 0	Non-folder		1980	509	133	3¼
Bison	37 0	46 0	37 0	17 6	3700	1256	108	3
COMMERCIAL MACHINES—								
Bristol ten-seater	41 8	54 3	Non-folder		4427	1788	115	4½
D.H. 34 six-seater	41 3	51 5	Non-folder		4453	1600	111	3½
D.H. 9 C three-seater	30 10	42 5	Non-folder		2456	800	110	5

(*a*) Vessels running on such a service from Great Britain to New York could send off aeroplanes which would reach Montreal before the steamer could get to New York. A corresponding gain would be made by Canadian steamers using the port of Halifax; on approaching Halifax the aeroplane could depart before the steamer reached the port, and *vice versâ*.

(*b*) Vessels running from Great Britain to Australia. These vessels could run to Port Said without a stop, and on their route, without stopping, could serve Lisbon, Gibraltar, Algiers, Malta, and Naples. They could then run from Suez to Colombo and deal with mails for Port Sudan, Khartoum, Aden, Somaliland, and possibly Bombay, while on the run from Colombo to Sydney, various other aerial services could be maintained.

(*c*) Vessels running from Vancouver to Hong Kong, or Australia, could in the same way deal with Pacific Islands and ports.

The design described in this paper was worked out in the first instance, and primarily, for a steamship serving as an aircraft carrier. The work has been completed on this basis, but it is considered that the resulting arrangements challenge very seriously those which have been adopted in the great majority of liners for the last thirty years.

On the usual plan the midship part of the ship is very largely taken up with funnels, uptakes, cowls, ventilating trunks, engine-room castings, &c. In the new arrangements these things are not to be found in the middle of the ship. The whole of the steam machinery is placed well aft; enormous clear decks are provided for passenger accommodation, and all the serious troubles due to overheating of passenger accommodation by the steam machinery are avoided. As electric cooking is proposed, there need be no fires or steam appliances forward or amidships except where steam appliances are fitted expressly for heating purposes.

The last thirty years have seen great developments in the machinery of liners. The reciprocating engine has given place to the steam turbine, and then to the geared steam turbine, while the internal combustion engine is also rapidly making its way. Radical improvements in steamship construction have followed on the lines, first, of the Isherwood system for longitudinal framing, and, secondly, on the straight-line system for transverse framing. But the design of the steamer herself, and the general arrangements, have only been developed in matters of detail.

The advantages of a ship designed after the style of that now suggested appear to be so great as to constitute a direct challenge to the present stereotyped arrangements for passenger liners as now built, quite apart from the question of special arrangements for carrying aircraft.—"The Engineer," March 23, 1923.

IS THE LIBERTY ENGINE "OBSOLESCENT?"

By L. D. Seymour.

The great aircraft engine designed by America's foremost engineers, developed and given to the military and naval forces of the United States in 1918 by the Army Air Service has neither been forgotten nor abandoned by the Army since the signing of the Armistice. With a large number of the engines available for future use at the close of the war, the Army has spared no effort to continue the unexcelled record of the Liberty.

Resulting from service use and exhaustive tests many refinements have been added and changes made. As new features of design or modification have proven of value, they have been incorporated in the engines in the possession of the Service and given not only to other branches including the Navy Department, but prominent manufacturers and designers for the general advancement of the art of aerial navigation.

While the characteristics of this engine are fairly well known to those connected with the Air Service, it may be found of value to recite some of the outstanding features for those not so intimate with aircraft. As originally designed with a weight of only 800 pounds, a horsepower of 400 was secured with the propeller shaft turning at approximately 1700 revolutions per minute. Two complete ignition systems were installed to guard against the possibility of trouble in the air from that source. Two duplex carbureters were used each divided into two units serving three of the twelve cylinders. Practically four separate carbureters were thus employed, but so set, adjusted and controlled that each cylinder would receive the same quality and quantity of fuel. A most interesting point in connection with the water system is the circulating pump. This pump though of the small centrifugal type is capable of delivering 100 gallons of water per minute with a free outlet. With the foregoing in view it is the more wonderful when compared with the ponderous proportions of even the lightest of steam power-plants installed in electric generating stations where even only a fraction of 400 horsepower is delivered. Not content with these characteristics, however, a constant effort has been made toward greater development with very gratifying results.

It is interesting to note some of the changes that have added immeasurably to the usefulness of an already marvelous powerplant.

In the following paragraphs note has been made of a few only of the changes which have occurred and the reasons therefor. From these one will be able to form an opinion of the work that has been done and what a part of the peace time duties of Uncle Sam's Air Service include.

THE PROBLEM OF LUBRICATION.

In an aircraft engine all working parts are usually lubricated by a direct oil lead carrying oil under pressure. Even though every means possible is employed to hold this pressure at the right value sometimes too much oil reaches the piston wall. This in the ordinary engine results in oil passing the piston, fouling spark plugs, forming carbon in the combustion chamber, hindering the proper operation of the valves, causing overheating, pre-ignition, etc. So that such a possibility might be reduced to a minimum, four small holes were drilled in the oil pressure relief valve giving an almost perfect balance to the oiling system at all speeds.

As an extra precaution, the pistons have been grooved and drilled so that excess oil is collected and drained back into the crank case or "sump" before it has had a chance of getting to the combustion chamber.

One of the greatest triumphs of McCook Field has been the development of a centrifugal oil cleaner. It has long been the practice of the Air Service to reclaim oil which has been used until impurities such as sediment, etc., have been collected, rendering it useless. This process while resulting in purification to the extent that the oil is of a better quality than the original, requires large tanks, heaters, etc., the weight of which runs into hundreds of pounds. It remained for Air Service engineers to develop the new centrifugal cleaner which is hardly bigger than one's two fists. This cleaner is made an integral part of the engine removing impurities of all kinds from the oil as fast as they are collected. This not only makes it simply necessary to add the oil that is actually consumed but makes draining and washing of the oil system a thing of the past.

Not only the engineer, but any layman can easily appreciate the importance of advantages accruing from these various devices such as: certain, constant, uniform oiling with never a "feast or a famine;" unheard of economy; prevention of carbon formation, cleaning seldom required and overheating from this source unknown.

FUEL SYSTEM AND FIRE HAZARD.

Naturally, in the air, fire is a greater danger than in almost any other place. For this reason no effort has been spared to reduce the chances of a fire starting from the power plant. The difficulties incident are very apparent when it is remembered that a 400 horsepower motor must be supported in the light frame of an airplane.

To this end flexible fuel line connections impervious to vibration have been developed. Carbureter air intake pipes where once gasoline vapor condensed and dripped back onto the engine have been led outside and above the engine housing to prevent the collection of gasoline where it could be ignited and cause damage.

By ingenious means and devices the engine's carbureters have been so changed that the gasoline consumption has been reduced approaching one-half the former amount. At the same time a much better proportion of gas and air have been secured which gives increased smoothness of operation, flexibility and complete combustion. One of the greatest advantages secured by these changes is the greater range of altitudes at which uniform operation is possible,

Probably the greatest cause of fires in the past has been the fact that gasoline was led to the carbureters under air pressure which, if a leak in the system occurred, caused the entire power plant, etc., to be subjected to a fine spray of fuel. This, of course, mixed with air, forms one of the most inflammable mixtures known. All this has been changed now with the successful development of a mechanically operated fuel pump supplying gasoline to the curbureter without the use of pressure in the tanks.

It is a well known fact that due to the rarification of the air at great heights, an internal combustion engine is able to deliver only a small fraction of its sea-level power. The supercharger, another newly developed accessory for the Liberty, removes this difficulty by delivering to the engine, at any explored altitude, air substantially the same as at sea level.

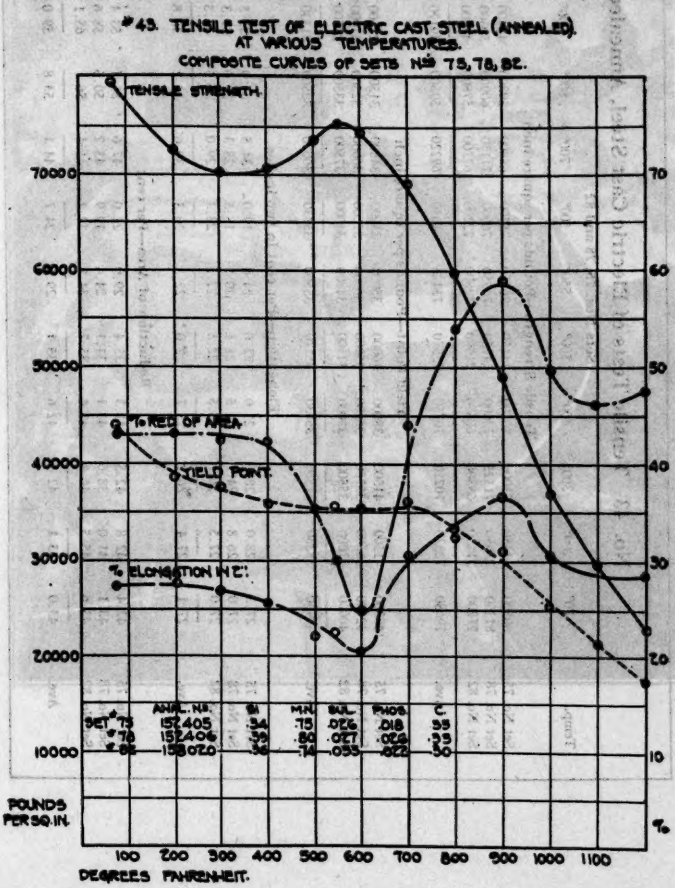
THE ELECTRIC SYSTEM.

Even though other parts and devices of any aircraft engine may be perfect the fact remains that the electric spark which fires the charge must be delivered at all times if there is to be any operation at all. In order to insure this many changes and modifications have appeared to guarantee non-failure from this source. Among other important items in redesign and reconstruction are the following: A 12 volt system has been substituted for the 8 volt, in addition allowing the employment of a self-starter; storage battery improvements; addition of buzzer distributor starters, safety relays, etc.; more completely armored cables and positive connections.

In other small features a recitation including redesigned flexible shafts, larger bearings, stronger gears, water system improvements, etc., could go on for many pages. However, those enumerated serve to show the untiring efforts that are constantly being exerted to keep the greatest war time engine also the greatest in peace. No attempt has been made to keep secret these discoveries which have added so much to our knowledge of engine construction in general, but rather they have been carefully explained and given to all interested in the progress of aeronautics including the Navy, Marine Corps, commercial concerns, etc. This attitude and action of the Army Air Service shows the value of this special research and will go a long way toward the development of the art in general. By such unselfish and patriotic endeavor will aerial navigation the sooner take its proper place in this country as the best means of transportation and communication. More than this, the same American spirit is shown to be alive that first gave to the world a practical solution of the problem of flight.—"Aerial Age," April, 1923.

TENSILE TEST OF ELECTRIC CAST STEEL.

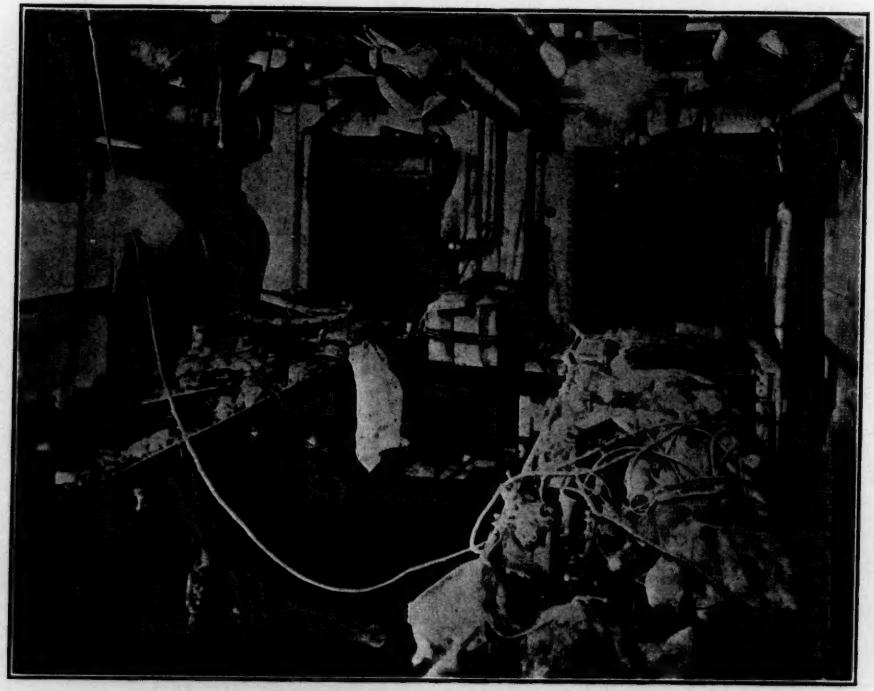
Within the last few months our testing laboratories have made very complete tests of our basic electric steels at high temperatures, extending the temperature range in the tests up to 1200 degrees F. As will be seen by the curve above and the figures on page 86 our present cast steel is as strong at 1200 degrees F. as our old steel was at 900 degrees F. and the yield point also carries the same relation. The new steel at 900 degrees F., which is incipient red heat, has a tensile strength of approximately fifty thousand pounds per square inch with a yield point of thirty thousand pounds. At 700 degrees F., the usual top of the super-heated steam temperature range, it has a tensile strength of sixty-nine thousand pounds and yield point of thirty-six thousand pounds per square inch. These figures and curves tell their own story.—"The Valve World," March, 1923.



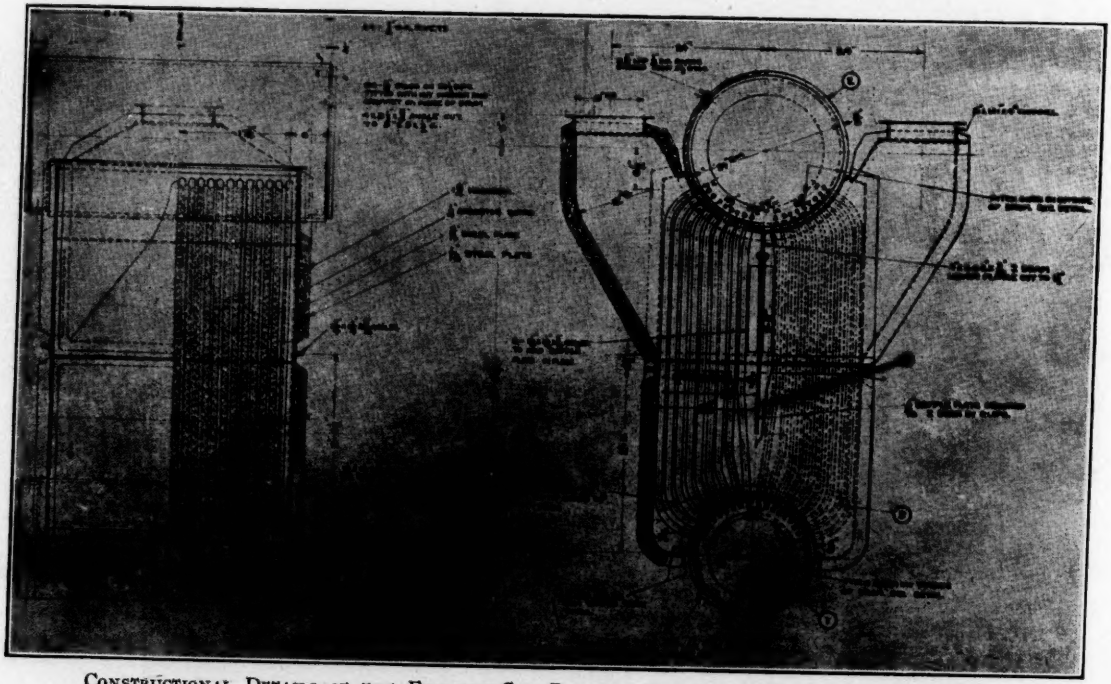
No. 43 Tensile Tests of Electric Cast Steel, Annealed

Sets Nos. 75, 78 and 82

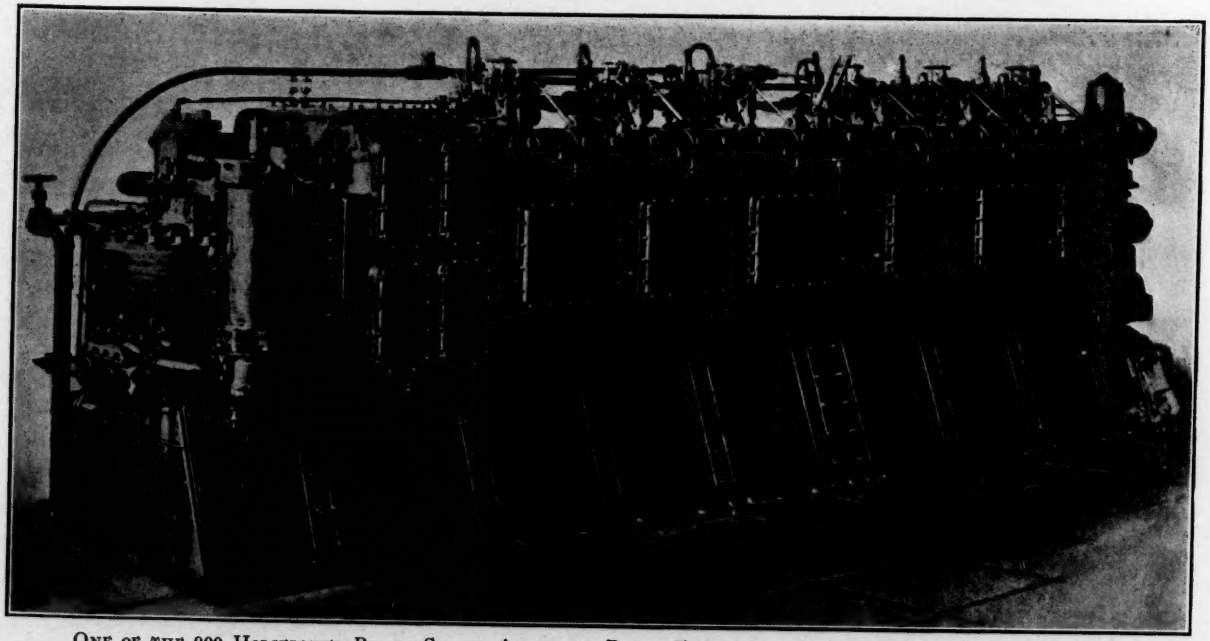
Temp.	70°	200°	300°	400°	500°	550°	600°	700°	800°	900°	1000°	1100°	1200°
Tensile Strength—Pounds per square inch													
Set No. 75	80500	73350	71000	70800	74450	76350	75700	68800	59800	48400	35900	26200	21700
Set No. 78	81350	73650	71325	72600	74750	76950	76000	71150	60900	50300	42450	32400	24400
Set No. 82	77200	71450	68300	68650	71500	72800	72200	67700	57800	48800	31750	29650	23100
Ave.	79680	72820	70210	70750	73570	75425	74630	69220	59500	49170	36700	29420	23070
Yield Point—Pounds per square inch													
Set No. 75	48300	41500	41500	38000	36000	35000	35000	34500	31500	28500	23000	17750	16900
Set No. 78	42000	36500	35750	36500	35000	36500	37000	36000	32500	37000	29500	25000	18500
Set No. 82	40000	37000	35500	33000	34500	34000	34000	37500	33500	28500	23000	21000	17000
Ave.	43430	38330	37580	35830	35170	35500	35330	36000	32500	31330	25170	21250	17470
Elongation—Per cent in two inches													
Set No. 75	27.3	28.0	28.0	27.0	22.0	24.0	19.0	34.5	35.0	35 5	29.5	28.7	27.5
Set No. 78	27.0	26.8	24.7	24.5	21 5	20.5	18.5	28.3	29.0	31.3	28.5	24.5	26.5
Set No. 82	27.0	27.5	26.0	25.5	22.5	22.5	23.3	29.0	35.5	43.5	34.0	32.0	31.5
Ave.	27.1	27.4	26.2	25.7	22.0	22.3	20.3	30.6	33.2	36.8	30,7	28.4	28.5
Reduction of Area—Per cent													
Set No. 75	42.4	42.8	42.5	41.3	35.4	29.9	22.0	47.6	54.7	53.4	46.4	48.4	48.8
Set No. 78	43.1	41.0	38.6	43.1	34.1	24.4	19.9	43.2	50.6	58.6	52.0	42.5	45.2
Set No. 82	43.6	45.5	46.4	43.4	35.7	34.8	32.2	41.4	56.1	65.1	50 4	47.5	48.4
Ave.	43.0	43.1	42.4	42.6	35.1	29.7	24.7	44.1	53.8	59 0	49.6	46.1	47.5



AUXILIARY-DIESEL ENGINE-ROOM ON U. S. S. "MARYLAND," WITH THE TWO 900 B.H.P. BUSCH-SULZER DIESEL ENGINES IN COURSE OF ERECTION. NOTE EXHAUST-GAS BOILERS IN THE BACKGROUND.



CONSTRUCTIONAL DETAILS OF THE EXHAUST-GAS BOILERS. COURTESY OF U. S. NAVY DEPARTMENT.



ONE OF THE 900 HORSEPOWER BUSCH-SULZER AUXILIARY DIESEL-ENGINES OF THE U. S. BATTLESHIP "MARYLAND."

DIESEL-ENGINES ON UNITED STATES BATTLESHIP.

FIRST PUBLISHED DETAILS OF INSTALLATION OF BUSCH-SULZER DIESEL-ENGINES, GENERAL ELECTRIC MOTORS AND WASTE-HEAT BOILERS ON THE DREADNOUGHT "MARYLAND."

Diesel-engines are becoming more numerous in naval craft as their advantages for certain uses are becoming recognized, several propelling-sets as well as ship's auxiliaries being now installed in the world's navies, not to mention the great fleets of submarines which are Diesel-propelled. However, one of the most interesting American naval installations we have seen is that being made on the Navy's greatest fighting machine, *Maryland*, which we were recently enabled to inspect through the courtesy and permission of Edwin Denby, Secretary of the Navy.

Not only does this plant include Diesel-engines and electric-generators, but the waste-heat of the exhaust-gases is utilized in an exceedingly interesting manner. The entire equipment is installed in a watertight compartment on the starboard side outboard and forward of the main engine-room, the only means of entrance or exit being up-and-over. There are no port-lights or other openings to the outside of the ship, but ventilators are arranged in connection with blowers. The chief purpose of the installation is to do away with the use of some of the ship's boiler equipment while in port, the Diesel-electric and distilling-set being much more economical than the boiler. However, the demands of steam for galleys, laundry and heating are now greater than a small Diesel set can supply. But the present installation will supply electric-current for lighting and power and steam for fuel-oil heating and for distilling fresh water.

Two six-cylinder Busch-Sulzer two-cycle Diesel-engines are installed in a compartment with the exhaust-heat boilers, the electric-generators being installed in another space forward of these and driven by a shaft extending through the bulkhead forward from each engine. Each engine will develop 900 brake horsepower at 350 R.P.M., but they will be operated at 300 R.P.M. to give 400 k.w. at the generator. When the installation is completed and put in operation Diesel-oil will be used for fuel for a half-hour upon starting and a half-hour before stopping, boiler-oil being regularly used while running. Eventually oil as low as 10 degrees Beaumé will be used—at least, we were given to understand. This, of course, is an exceptionally heavy oil. Daily service tanks of 150 and 300 gallons are provided on the forward bulkhead of the engine-compartment.

At the aft end of this space, as illustrated in the accompanying view photographed by permission of Secretary Denby before completion of the work a few days before the *Maryland* put to sea, are two exhaust-gas boilers, one for each Diesel-engine. These were designed by Commander C. A. Jones of the Bureau of Engineering, Navy Department, through whose courtesy we are enabled to publish drawings of these interesting units. Each of these boilers contains 424 tubes of 1-inch outside diameter, 95 mils. thick and has a heating-surface for the tubes only of 445 square feet, these tubes being of seamless cold-drawn steel. Provision is made for cleaning the tube surface in the event of deposits from the waste-gases. The test pressure for the boilers is 50 pounds per square inch and the drums are open hearth medium steel. Insulation consists of carbonate of magnesia containing about 8 per cent asbestos fiber.

It is expected that an average of about 400 to 500 pounds of steam per hour will be generated at the average load condition which the generator will carry when producing light and power for the ship. The exhaust-line of the engine is fitted with arrangements for by-passing the boiler so that

in case of necessity the engine's operation need not be interfered with during overhaul or cleaning of the boiler. Feed for the boiler comes from the regular fresh-water boiler-feed system of the ship. Steam will be generated for the primary purpose of obtaining steam at 10 pounds pressure to be used for heating the fuel-oil piping and any excess of steam to be utilized in the ship's distilling-plant for the production of fresh-water.

The electrical equipment is by the General Electric Co. and the generators are three-wired, equipped with balancing sets for voltage regulation. Double-throw switches are provided to give 120 volts for lighting circuit and 240 volts for power. This installation will produce sufficient current to take considerable load off the ship's steam-plant.—"Motorship," April, 1923.

VARIABLE-SPEED ALTERNATING-CURRENT MOTORS WITHOUT COMMUTATORS.*

Without ignoring the work of early inventors, such as Dahlander and Lindström and Alexanderson, who have provided means of adapting a single winding to give, for instance, two numbers of poles having a fixed ratio, it is not unfair to say that the idea of using the method of pole changing to produce a machine having anything resembling a gradual variation of speed by small steps between wide limits, would have been regarded a few years ago as chimerical. Yet it is also obvious that if by any means we could render practically available all the numbers of poles which exist within a given range, we would have a machine with a very high degree of practical usefulness. For instance, on fifty periods between the speeds of 375 revolutions per minute and 1000 revolutions per minute there are six numbers of poles available.

One method by which all these numbers of poles might theoretically be rendered available with a single winding has long been known, *i.e.*, the use of a ring winding tapped at the various points necessary to give different numbers of poles—as shown in Fig. 1. It is well known that a ring winding can be used on any number of poles, if a number of equidistant tappings equal to three times the number of pairs of poles are brought out on each speed, equalizer connections being made between tappings 1 to 4 throughout the circumference. To obtain six numbers of poles from the same winding in this way, however, would require a formidable number of tappings, and to connect them together in the various manners required would involve an equally formidable type of switchgear. Various other difficulties are also met with.

The solution to all these difficulties was only found gradually. In order to overcome the necessity for a very large number of terminals, a method of pole changing entirely different from that described above was adopted. This method is shown in Figs. 2 and 3. Still considering a ring-wound machine, assume it to be divided into twelve sections, shown developed in Fig. 2.

At this point it may be convenient to describe a diagrammatic method of denoting the windings of alternating-current machines which I have developed in order to simplify winding diagrams. It will be seen that the ring winding shown at the top of Fig. 2 is divided into twelve sections, each section being shown as consisting of two turns. The beginning of each section is denoted by a white circle and the end by a smaller blackened circle, the extremities of the various sections being connected in series in the usual

* Abstract of paper read before the Institution of Electrical Engineers by Mr. F. Creedy.

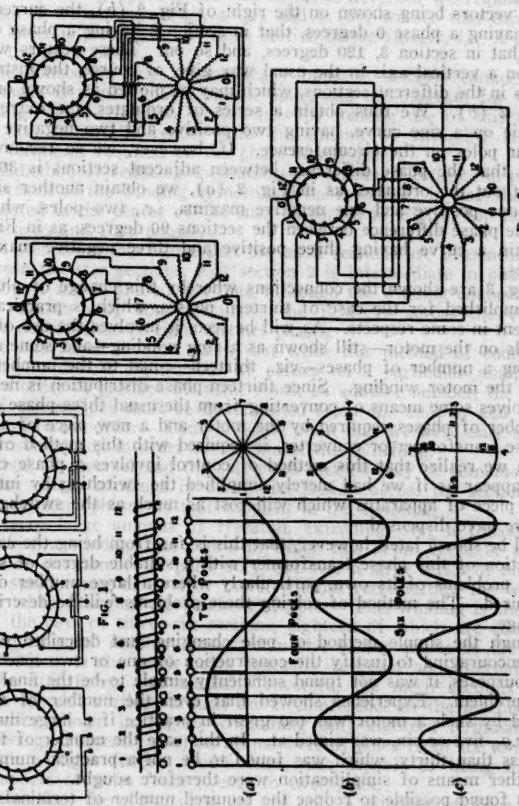


Fig. 3.
Fig. 1.
Two Poles.
Four Poles.
Six Poles.
(a)
(b)
(c)
Fig. 2.

manner. An obvious simplification for diagrammatic purposes is to omit the drawing of the winding and merely retain the two terminals of each section, joining them by a straight line in order to show that they belong to th esame section. This diagrammatic method of denoting the winding is shown immediately below the drawing of the ring in Fig. 2, the sections being numbered in order round the circumference, and the method is made considerable use of in these notes.

Imagine the twelve sections of the ring winding shown in Fig. 2 to be carrying currents differing in phase from one another by 60 degrees, the current vectors being shown on the right of Fig. 2 (*b*), the current in section 1 having a phase 0 degrees, that in section 2 having a phase of 60 degrees, that in section 3, 120 degrees, and so on. These vectors when projected on a vertical axis in the usual way give, of course, the instantaneous currents in the different sections, which may be plotted as shown on the left of Fig. 2 (*b*). We thus obtain a series of ordinates, the extremities of which lie on a sine curve, having two positive and two negative maxima, *i.e.*, four poles on the circumference. If, however, we so reconnect these sections that the phase difference between adjacent sections is 30 degrees, and plot out the ordinates as in Fig. 2 (*a*), we obtain another sine curve having one positive and one negative maxima, *i.e.*, two poles, while if we make the phase difference between the sections 90 degrees, as in Fig. 2 (*c*), we obtain a curve having three positive and three negative maxima, *i.e.*, six poles.

In Fig. 3 are shown the connections whereby this change of phases may be accomplished for the case of thirteen phases, which is practically more convenient in some respects. As will be seen, it involves the use of thirteen terminals on the motor—still shown as a ring winding—and some means of producing a number of phases—viz., thirteen—equal to the number of sections in the motor winding. Since thirteen-phase distribution is never used, this involves some means of converting from the usual three-phase supply to the number of phases required by the motor and a new piece of apparatus, the phase transformer or converter, is required with this method of control.

When we realize that this method of control involves a phase converter, it may appear as if we had merely simplified the switching by introducing another piece of apparatus which will cost as much as the switchgear with which we have dispensed.

It will be shown later, however, that this is far from being the case. The construction of this phase transformer with a suitable degree of simplicity involves problems of its own, particularly where a large number of phases are required. The method of solving these problems will be described at a later stage.

Although the simple method of pole changing just described was sufficiently encouraging to justify the construction of one or two machines for special purposes, it was not found sufficiently simple to be the final solution of the problem. Experience showed that even the number of terminals possessed by such a motor was too great in practice if a large number of speeds, *e.g.*, five or six, was aimed at. In this case the number of terminals is not less than thirty, which was found to be not a practical number, and still further means of simplification were therefore sought.

It was found possible to reduce the required number of terminals in several ways. It is clear, first, that wherever we have currents differing 180 degrees in phase, that is to say, mutually reversed, it is not necessary to supply two distinct phase-transformer terminals to obtain these two phases. If the two sections are connected in series or parallel, mutually reversed, they can be supplied from a single terminal.

For instance, returning to Fig. 2, if we wish to connect the motor only for two poles and six poles we find that in both these cases sections 1 and 7, 2 and 8, &c., are diametrically opposite in phase. We may, therefore, connect them in series reversed, as shown in Fig. 4 (*b*), in which the sections are denoted by the diagrammatic method already described, and the direction of the straight line joining the two extremities is made to represent the phase of the current in the section.

Connecting all the sections in star, we require twelve terminals to give two poles. Connecting opposite sections in series reversed we require only six, although if we compare Figs. 4 (*a*) and 4 (*b*), we see that each section in the latter is parallel to the same section in the former, *i.e.*, it carries current of the same phase.

The same connection—viz., diametrically opposite sections connected in series reversed—is also available on six poles, and the diagram for this is shown in Fig. 4 (*c*). Hence, we see how by making use of this simple device we may materially reduce both the amount of switching and the number of terminals required on the phase transformer.

A second method of reducing the number of terminals consists in putting alternate sections of the winding in mesh and the remainder in star, the mesh being connected in the center of the star, as shown in Fig. 5, for the case of eight sections and four phases.

It will be seen from this figure that the mesh sections are intermediate in phase between the star sections, *e.g.*, section 2 is intermediate in phase between sections 1 and 3, and so on; thus, although the winding connected, as in Fig. 5, requires only four terminals, the E.M.F.'s across the various sections are nevertheless in eight distinct phases.

Again, it was found in most cases that while this method of connection might suffice to double the number of phases on a particular number of poles, yet in order to produce the same result on another number, the whole or many of the sections must be disconnected from one another and reconnected in a different order. Thus it would seem that we have only simplified the transformation in order to complicate the switching.

The method of reversal and the star-mesh method may be combined, and thus we may obtain windings in which the E.M.F. across the sections has four times as many phases as that supplied to the terminals.

After a very large amount of research, extending over a considerable period of time, a winding was found in which this result could be accomplished, not on one number of poles only, but over a large series of numbers covering a speed range of 3—1 or more, without reconnecting the winding in any way when changing the numbers of poles. This winding is shown in Fig. 6 for the case of thirty-six sections, eighteen of which are connected in mesh and an equal number connected in star to the angles of the mesh, alternate sections in the mesh being mutually reversed and alternate sections in the star being also mutually reversed, the mesh sections alternating with the star sections around the circumference of the machine.

The winding shown in Fig. 6—containing only eighteen terminals—can be, and is being, frequently used on all numbers of poles from six to sixteen and is, with a slight further modification, also capable of being used on further numbers of poles from twenty to twenty-six.

This connection, therefore, gives us not only a winding of very simple character with a moderate number of terminals—eighteen—capable of reconnection in a simple manner for a large number of different numbers of poles, but it also gives us the characteristic which was shown above to be desirable, viz., that of having an approximately constant flux in all numbers of poles. It is difficult to explain the cause of this in a simple manner, but one may say briefly that it is due to the fact that the number of

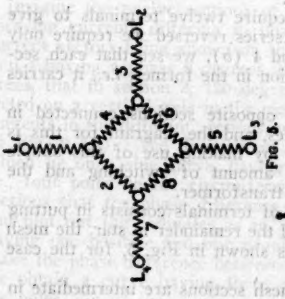


Fig. 5.

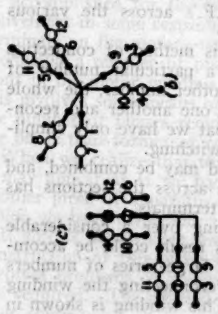
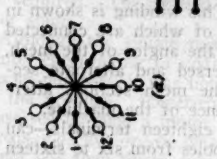
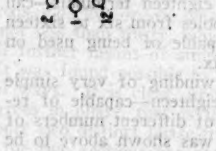


Fig. 4.





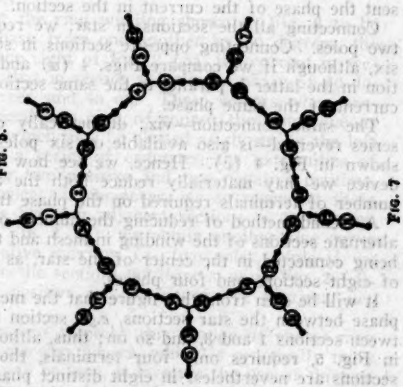


Fig. 7

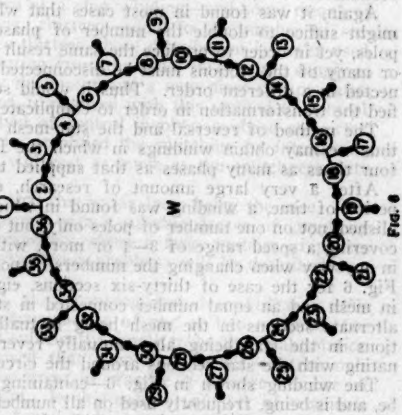


Fig. 8

sections in series between terminals having a given phase difference of, say, 120 degrees, becomes gradually greater and greater as we increase the number of poles, or decrease the speed, so that as the rate of rotation of the flux wave gets less and less, due to the decreasing speed of the motor, the voltage across any section having a given number of turns also becomes less and less approximately in the same proportion, and thus about the same flux is needed to balance this voltage at all speeds.

A further simplification is possible where, for instance, only odd numbers of pairs of poles are required and, consequently, diametrically opposite sections are always opposite in phase on every number of poles which a machine is required to give, and, therefore, may be permanently connected in series, mutually reversed.

The winding of such a machine as shown in Fig. 7 and, as will be seen, has only nine terminals, capable of operating on two, six, ten and fourteen poles, and, with a slight further modification, on twenty-two and twenty-six poles. This winding is, I believe, the simplest known multi-speed winding. It requires no more than nine phases, while the original winding may require eighteen phases.

THE PHASE CONVERTER.

Having described how the difficulties connected with designing a multi-speed winding have been overcome by means involving the use of a phase transformer or converter, it will next be desirable to describe how this apparatus is constructed in practice. In certain cases, *e.g.*, where a single motor is operated from a generator which serves no other purpose, as in ship propulsion, no such transformer is needed, as there is no difficulty in winding the generator for the number of phases required; but in the majority of cases the machine, if it is to be of any practical use whatever, must operate from a power supply having, say, not more than three phases, and hence it is of vital importance to the practical success of the apparatus that this phase transformer should be not merely technically satisfactory, but should also be capable of being built at a very moderate price.

By restricting the number of phases to multiples of three, the transformers to be excited from a three-phase circuit may be simplified very materially, and it will be convenient to describe the transformer as used in connection with the winding previously referred to. This transformer does not differ externally in any way from the standard three-phase core-type transformer, built in the usual way with three exactly similar limbs. The connections, however, are as shown in Fig. 8, in which all the windings shown by lines parallel to one another are considered to be wound on the same limb of the transformer. Since the windings shown in this figure are drawn parallel to three straight lines at 120 degrees, it follows that only three such limbs are needed.

If we take a three-phase transformer having two secondary sections on each limb, and connect these sections in the manner shown, which may be called a hexagon connection, we obtain an apparatus from which six-phase currents may be derived by bringing out terminals from the corners of the hexagon. If, however, we tap the sides of the hexagon in the manner shown, winding on the three limbs, in addition, six small, auxiliary sections to be connected to the corners of the hexagon, it is possible without further elaboration to obtain no less than eighteen phases, as has been amply proved in practice, the tappings shown in the diagram lying in a circle.

So far, we have regarded the hexagon winding with tappings as being the secondary, and have assumed that there is a primary winding which carries the three-phase currents. But if no voltage transformation is needed, as in this case, a separate primary is not needed, for the primary

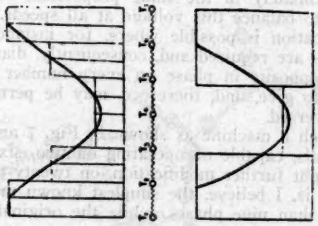


FIG. 10.

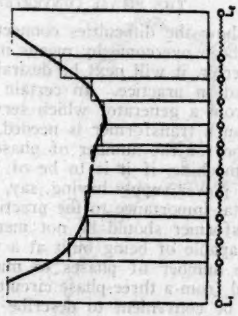


FIG. 9.

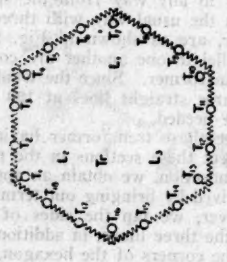


FIG. 8.

terminals may be connected direct to alternate corners of the hexagon, or even in order to save extra terminals, to the extremities L_1, L_2, L_3 of three of the small auxiliary sections wound on the transformer and corresponding to the secondary terminals T_1, T_7, T_{13}. This latter plan is adopted to avoid the necessity of bringing out three extra primary terminals.

By using the apparatus as an auto-transformer in this way we at once abolish the whole of the primary windings and, therefore, reduce the amount of copper on the transformer to half, or, in other words, double the rating of a transformer of a given size.

We do more than this, in fact, because the primary and secondary currents flowing in opposite directions in the same windings give a resultant which is very materially less than the secondary current alone. This is exemplified in Fig. 9, which shows the currents in the sections of the thirty-phase transformer, which formed part of the original experimental machine, which was constructed to test the present method of speed variation in its earliest form, L_1, L_2 being the three-phase terminals.

It will be seen that the current in the neighborhood of the three-phase terminals is much greater than at points intermediate between them, and, in fact, falls off to less than one-half at a point midway between the terminals.

Fig. 10 shows a similar curve for up-to-date nine-phase and eighteen-phase transformers as now being constructed commercially by the Metropolitan-Vickers Electrical Company for use with these machines. The same reduction in current per section at a point intermediate between the three-phase terminals T_1, T_7 is noticed here, and the analogy between the reduction of current in these transformers and the similar reduction which takes place in the armature of a rotary converter in sections intermediate between the tapping points may be referred to.*

As a result of this further reduction a phase transformer to transform a given amount of power, say, from three to nine or eighteen phases, will require a size of transformer of only between 30 to 40 per cent of the rating of the transformer which will be required to convert the same amount of power, say, from one voltage to another.

Still another economy which has the effect of still further reducing their capacity can be effected in the use of these phase converters. In a constant-torque motor giving a horsepower proportional to the speed, the amount of power taken will also be approximately proportional to the speed. For instance, the power taken at 750 revolutions per minute will be only 0.75 times the power taken at 1000 revolutions per minute, and if we can arrange the winding of our motor so that it is directly connected to the line when operating at 1000 revolutions per minute, the phase converter not being used on that speed but coming into operation for the first time only at 750 revolutions per minute, we shall be able to reduce its capacity to 0.75 times the value which would be needed to supply the power required at 1000 revolutions per minute. By way of analogy to this plan, we may refer to an automobile gear box in which it is usual to give direct drive on the top speed, thereby reducing the amount of power to be transmitted by the gears. The above is the electrical form of the direct drive.

As a result of the above construction and method of utilizing the phase transformer, the rating of such a transformer may be only from 18 to 22 per cent of the transformer rating needed to handle the maximum power taken by the motor at top speed. It forms, therefore, by no means an expensive item in the equipment, much less so, in fact, than the transformer

* I hope to publish the full theory of these phase converters on another occasion.

required with many types of variable-speed commutator motor, the capacity of which is proportional to the difference between the speed of the machines and synchronous speed.

The remainder of the paper describes the controller and deals with the characteristics of the equipment. There is also a section on the cascade motor, designed by Mr. L. J. Hunt.—"The Engineer," Jan. 12, 1923.

BOOK REVIEW.

POSITION FINDING AT SEA, BASED ON THE METHOD OF DE SQUINO, BY GILBERT R. CHASE, LIEUTENANT COMMANDER, U. S. N., RETIRED. (D. Van Nostrand Co., 8 Warren St., N. Y.)

A very compact exposition of position finding by the use of Squino tables. The author makes no claim to any contribution to the theoretical side of navigation, but only to a concise illustration of the method of using these tables. The book should be very useful to those beginning the study of practical navigation.

ASSOCIATION NOTES.

The Society held its annual banquet on February 27, 1923, at the New Willard Hotel, Washington, D. C. There were present 178 members and guests.

The occasion was notable for the unusual number of distinguished speakers and the interesting subjects that were covered. They were as follows:

"The Navy as a National Asset,"
THE SECRETARY OF THE NAVY.

"Standardization and Simplification of Engineering Practice,"
THE SECRETARY OF COMMERCE.

"The Bureau of the Budget,"
GENERAL H. M. LORD, U. S. A., DIRECTOR OF THE BUDGET.

"The Fleet,"
ADMIRAL R. E. COONTZ, U. S. N., CHIEF OF NAVAL OPERATIONS.

It seemed to be the consensus of opinion of those present that the banquet was one of the most successful that has ever been held. Commander F. J. Cleary performed the duties of toastmaster in the able manner which he always displays on occasions of this character.

The success of the banquet was entirely due to the efforts of the banquet committee which was composed of Commander F. J. Cleary, U. S. N., Commander H. S. Howard, (CC) U. S. N., and Commander O. L. Cox, U. S. N. They were tendered a vote of thanks by the Council.

The attention of all members is called to the fact that the Society is now making an attempt to enroll all officers as members and it is requested that every member lend all possible assistance in furthering this project by informing other officers as to the purposes of the Society and the value of membership in it.

The following members have joined the Society since the publication of the last number of the JOURNAL:

NAVAL.

Barnett, J. Walter, ex-Lieut., 208 Evans Bldg, Washington, D. C.

Griswold, W. E., ex-Lieut., 21 McDougal Alley, New York City.

McLean, Ridley, Captain, U. S. N.

Patterson, Gordon A., Ensign, U. S. N.

Stubbs, Hubert K., Lieutenant, U. S. N.

White, Henry E., Lieutenant, U. S. N.

CIVIL.

Colston, Robert, 90 West St., New York City.

de Leon, Carlos, Troy Club, Troy, N. Y.

Jefferson, Carl J., 326 Horace Avenue, Palmyra, N. J.

Meyer, Henry C. E., care Gibbs Bros., Inc., 1 Broadway, New York City.

Watson, Walter W., 22 Spear St., Quincy, Mass.

ASSOCIATE.

Tode, Arthur M., Lieut. N. R. F., New York Nautical Training Ship *Newport*, care Postmaster, New York City.

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